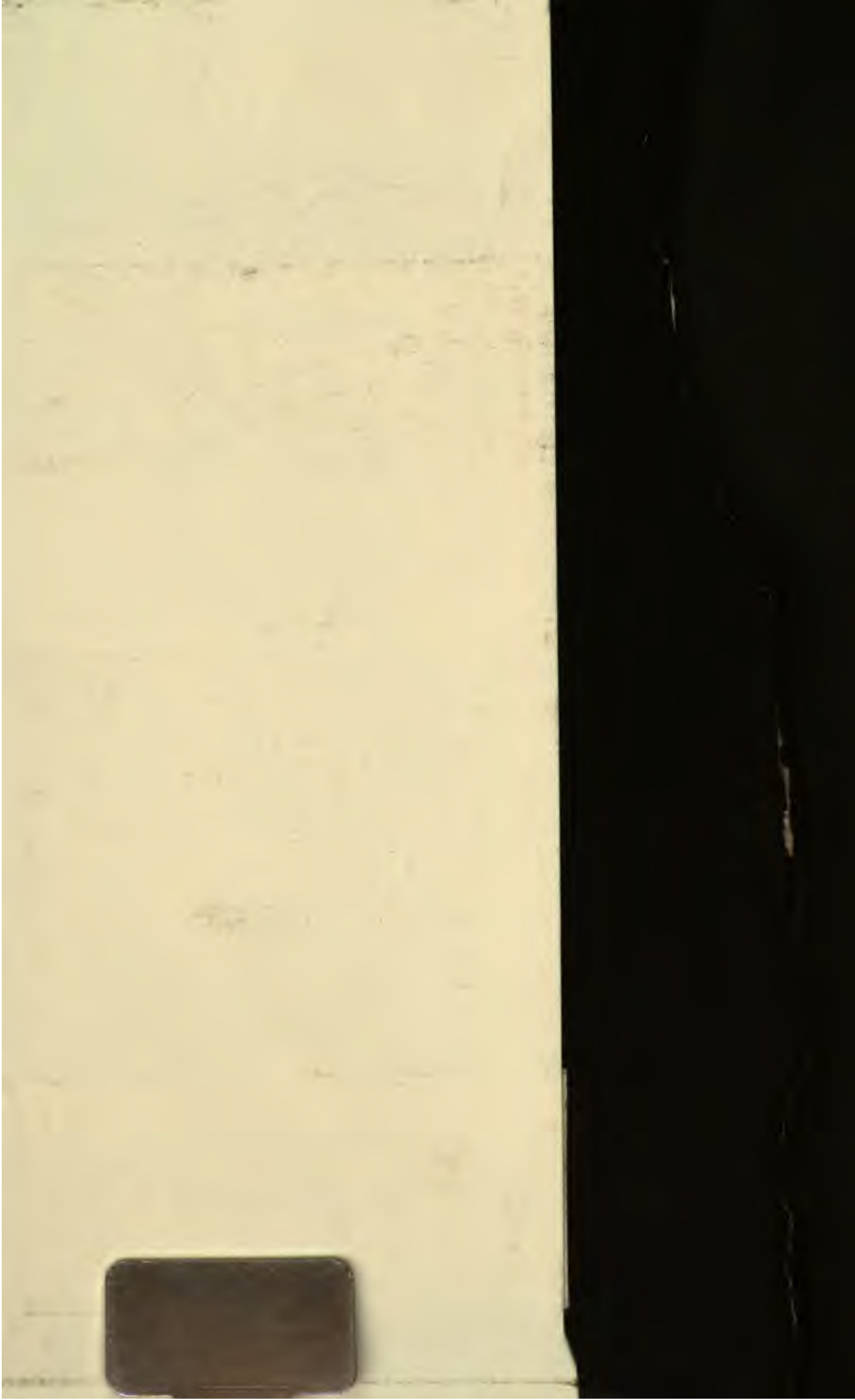

THE

MARINE STEAM ENGINE

PRINTED BY
SPOTTISWOODE AND CO., NEW-STREET SQUARE
LONDON

THE

MARINE STEAM ENGINE

A TREATISE FOR THE USE OF ENGINEERING STUDENTS
AND OFFICERS OF THE ROYAL NAVY

BY

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WITH NUMEROUS DIAGRAMS

SECOND EDITION

LONDON
LONGMANS, GREEN, AND CO.
1885

To

ADMIRAL THE RIGHT HON.

SIR ASTLEY COOPER KEY, G.C.B., F.R.S., &c.

FIRST AND PRINCIPAL NAVAL AIDE-DE-CAMP TO THE QUEEN

AND LATE FIRST NAVAL LORD OF THE ADMIRALTY

This Book

IS RESPECTFULLY DEDICATED BY

HIS MOST OBEDIENT SERVANT

THE AUTHOR

NOTE TO SECOND EDITION.

THE demand for a Second Edition came somewhat unexpectedly, and owing to the pressure of my official duties I have been unable to revise the work to any great extent. Information has, however, been added relating to the more recent developments of marine engineering, more especially with reference to triple expansion engines and closed stoke-holds with forced draught, which are now generally adopted for war-ships. My thanks are due to the reviewers of the First Edition who so kindly indicated points in which modifications appeared to be desirable, and their remarks have, so far as possible, been kept in view in preparing the present edition. In the revision for the press and the correction of proofs, much assistance has been kindly rendered by Mr. T. Soper, R.N., Engineer Inspector at the Admiralty.

RICHARD SENNETT.

SURBITON, 1885.

PREFACE.

My principal object in preparing this Treatise has been to provide a text-book suitable for the use of Students of Marine Engineering, and of Naval Officers who are desirous of obtaining a general knowledge of the machinery of ships which they may have to command.

During my period of service as Instructor and Lecturer in Marine Engineering at the Royal Naval College, and also since that time in instructing the Engineer Students in H.M's Dockyard, I have frequently experienced great difficulty in recommending any suitable text-book to the students; most of the existing works on the steam-engine being either too general in scope or theoretical in character.

I have not considered it desirable to devote much space to the discussion of the early history of Engineering, nor to the description of what are now obsolete types of machinery, which, though interesting to those who have

sufficient time and leisure for the study, would be of little value to the class of students for whom this book is mainly designed. For the most part I have preferred to confine attention to the more recent examples of Marine Engineering practice, and have endeavoured to explain, as clearly and briefly as possible, the principles on which the marine steam-engine should be constructed and worked. The theoretical parts of the subject have been treated throughout the book in popular language, the use of mathematics having been avoided so far as possible.

The opening chapter contains a brief sketch of the progress that has been made in Marine Engineering, more especially during the last 25 to 30 years, from which it will be seen that although we may not have attained to so high a state of efficiency as could be desired, yet great strides have been made, and the record will not appear on the whole unsatisfactory.

From the nature of this book there cannot be much claim to originality except in arrangement, and the sources from which information has been gained have been acknowledged as far as possible in the text. My principal aim has been to place ascertained facts in such a form that they may be readily apprehended by students of Marine Engineering, and thus to supply a want that has long been felt.

In preparing the chapters on the properties of steam, much assistance has been obtained from the valuable

treatise of Professor Cotterill on the steam-engine, and it is almost needless to state that no work of this character could be prepared without drawing largely from the scientific manuals and papers of the late Professor Rankine, who may justly be considered as the founder of the modern theory of the steam-engine.

In the revision for the press, much assistance has been kindly rendered by my colleague, Mr. W. W. Chilcott, F.R.S.N.A., &c., Chief Engineer, Royal Navy; and second assistant to the chief engineer of Devonport Dockyard.

RICHARD SENNETT.

Devonport, 1882.

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THE

MARINE STEAM-ENGINE.

PART I.

INTRODUCTORY.

CHAPTER I.

EARLY HISTORY AND PROGRESS.

THE earliest steam-engines were simply reciprocating engines, and it was not until a suitable method of turning reciprocating into rotative motion had been discovered and utilised that any progress was, or could be, made in adapting the steam-engine to the propulsion of vessels. The adoption of the crank, which enabled the power of the engine to be transmitted to the propeller smoothly and without shock, was an indispensable step in the progress of steam navigation.

The marine steam-engine may justly be considered as a production of the present century. In the latter part of the eighteenth century several crude attempts were made to adapt the steam-engine for the propulsion of boats, but none of them were quite successful. The first practical steamboat was built on the Clyde, in 1801, by William Symington for Lord Dundas. She was called

the 'Charlotte Dundas,' and was worked for some time with success as a tug on the Forth and Clyde Canal, but was withdrawn from this service in consequence of an apprehension that the banks of the canal would suffer from the wash of the propeller. This boat was fitted with a single paddle-wheel placed near the stern, driven by a horizontal, direct-acting engine, with connecting-rod and crank, and the general arrangement of her machinery would be considered creditable even at the present day.

The first recorded instance of steam navigation proving commercially successful was in America, where, in 1807, Robert Fulton built a steam-vessel called the 'Clermont,' propelled by paddles driven by a Boulton and Watt engine. In 1812, Henry Bell built a vessel called the 'Comet,' which was successfully worked on the Clyde as a passenger steamer between Glasgow and Greenock. The 'Comet' was propelled by two pairs of paddles, each paddle having four floats or blades, somewhat resembling a pair of canoe paddles, crossed at right angles. The paddles were driven by an engine of somewhat peculiar design, which, however, approximated to the side-lever engine of a later day. This boat, which was only 40 ft. long and 10½ ft. broad, was the first passenger steamer in Europe.

From this date the success of steam navigation may be said to have been secured, and the advancement that has been made since has not consisted so much in the discovery of new principles as in the introduction and development of improved mechanism and workmanship, with consequent economy of fuel. The result has been a progressive increase in the size, power, and speed of steam-ships and in the extent of their voyages; so that at the present day we have ships displacing more than 10,000 tons, and capable of being driven at speeds of sixteen to eighteen knots per hour by engines developing more than 10,000 indicated horse power.

Side-lever engine.—The propeller used in the earlier steamships was invariably the paddle-wheel, and the type of engine generally employed was that known as the side-lever engine, which may be considered as the marine counterpart of the beam engine so universally used at that time for land purposes.

The general arrangement of the side-lever engine is shown in outline in Fig. 1.

On the top of the piston-rod is fixed a crosshead with side rods, S, attached at each end, which, passing down on either side of the cylinder, are connected to the ends, A, of a beam or side-lever, A B, oscillating on a fulcrum or gudgeon at its centre, C. The opposite ends, B, of these side levers are fitted with journals carrying the crosstail, to the centre of which, one end of the connecting-rod B D is attached, the other end working on the crankpin D. The air-pump E is worked by side rods from intermediate points in the side-levers, the upper ends of the air-pump side rods being jointed to the opposite ends of the air-pump crosshead, to the centre of which the air-pump rod is secured. The piston-rod crosshead worked in vertical guides to insure parallelism, and the parallel-motion rods which are necessary in land beam engines were dispensed with.

The arrangement of the side-levers was sometimes varied by making them levers of the third order, the gudgeon or fulcrum being at one end and the steam cylinder placed between the gudgeon and connecting-rod. These engines were commonly known as *grasshopper engines*.

The side-lever type of engine, though very heavy and occupying a large space for the power developed, was safe and reliable, and consequently continued in general use for a great number of years, but it was at length superseded by direct-acting engines, which were lighter and more compact.

Steam vessels were introduced into the Royal Navy

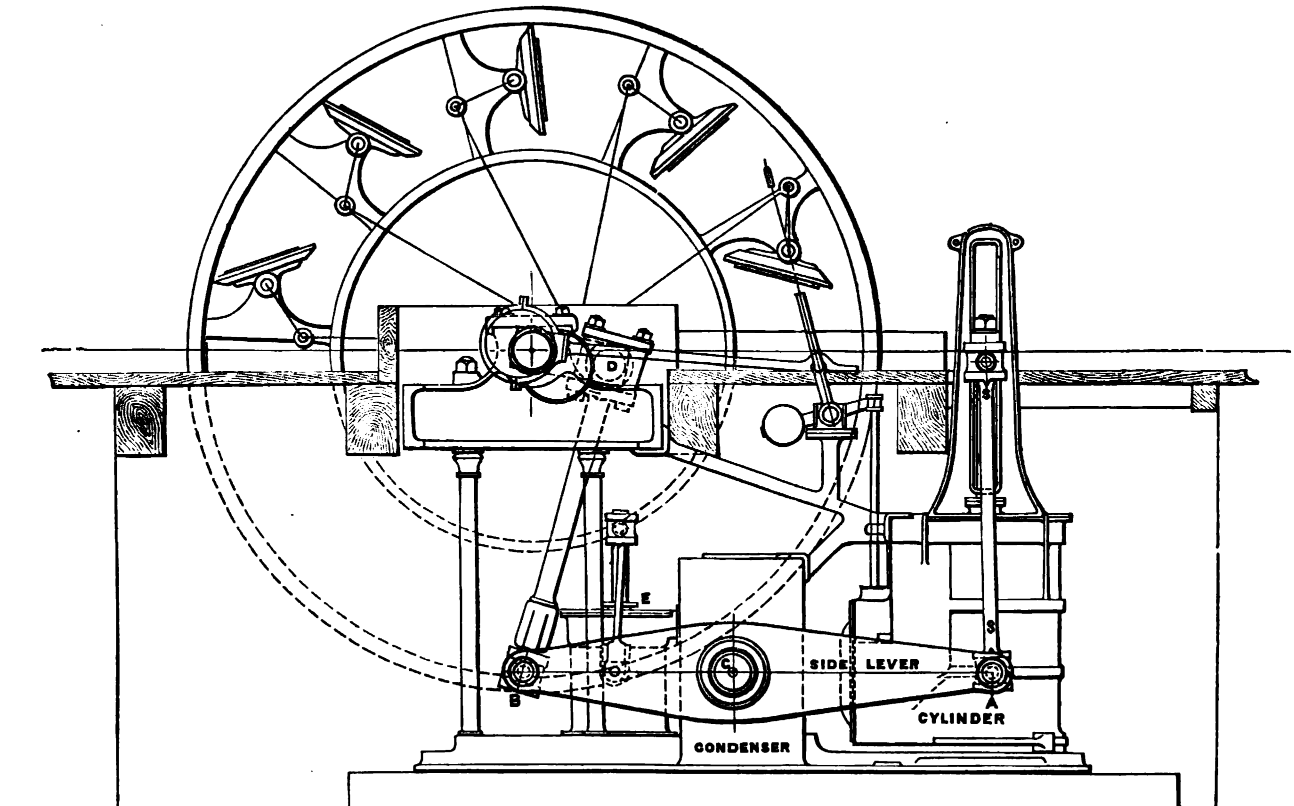

Fig. 1.

in the year 1827. On December 4th in that year three small paddle-wheel steamers, called respectively the 'Echo,' 'Lightning,' and 'Meteor,' were commissioned, and their names appeared for the first time in the Official Navy List for March 1828. The engines of these boats were each of 100 horse power nominal. They were commanded by naval lieutenants, but were mainly used for towing and general purposes, and can scarcely be classed as war vessels. Between this date and 1840 seventy other steam vessels were added to the Navy, the majority being fitted with flue boilers and heavy, slow-moving side-lever engines worked with steam at a pressure of 4 lbs. per square inch above the atmosphere.

The 'Rhadamanthus' was one of the steam-ships added to the Navy during that period. She was fitted with side-lever engines and flue boilers by Messrs. Maudslay, Sons & Field in 1832. The nominal horse power was 220, but the engines were capable of being worked up to 400 I.H.P., or 1·8 times the nominal power. The load on the safety valves was 4 lbs. per square inch, and the number of revolutions per minute when working at full power 17½, giving a piston speed of 175 ft. per minute. The total weight of the machinery was 275 tons, or 13·75 cwts. per I.H.P. developed.

Between 1840 and 1850 tubular boilers were introduced. In these boilers a group of small tubes was substituted for the long winding flue to convey the heated gases from the furnaces to the uptakes. The boilers were thus made lighter and more compact, and the working pressures of steam generally were increased to from 10 to 15 lbs. per square inch above the atmosphere. The space and weight occupied by the engines were also reduced by improved designs and the use of higher speeds of piston, but the majority of steam vessels afloat were still propelled by paddle-wheels.

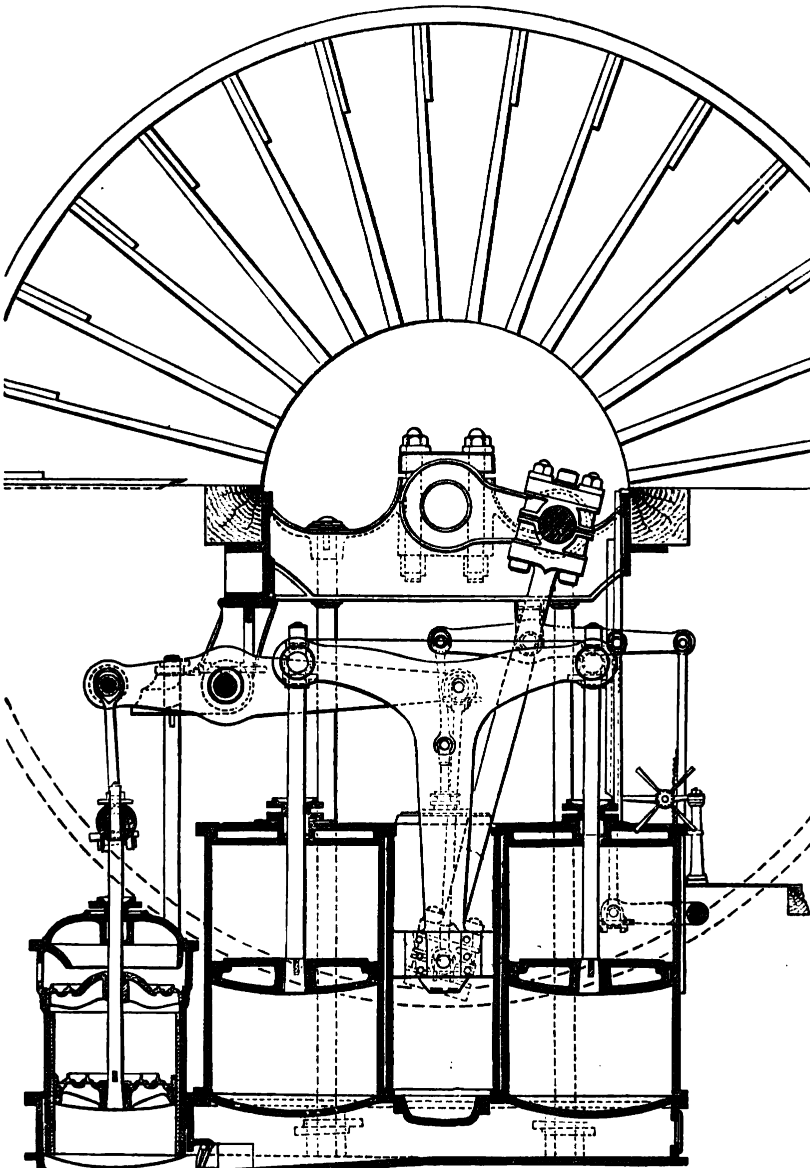

Fig. 2.

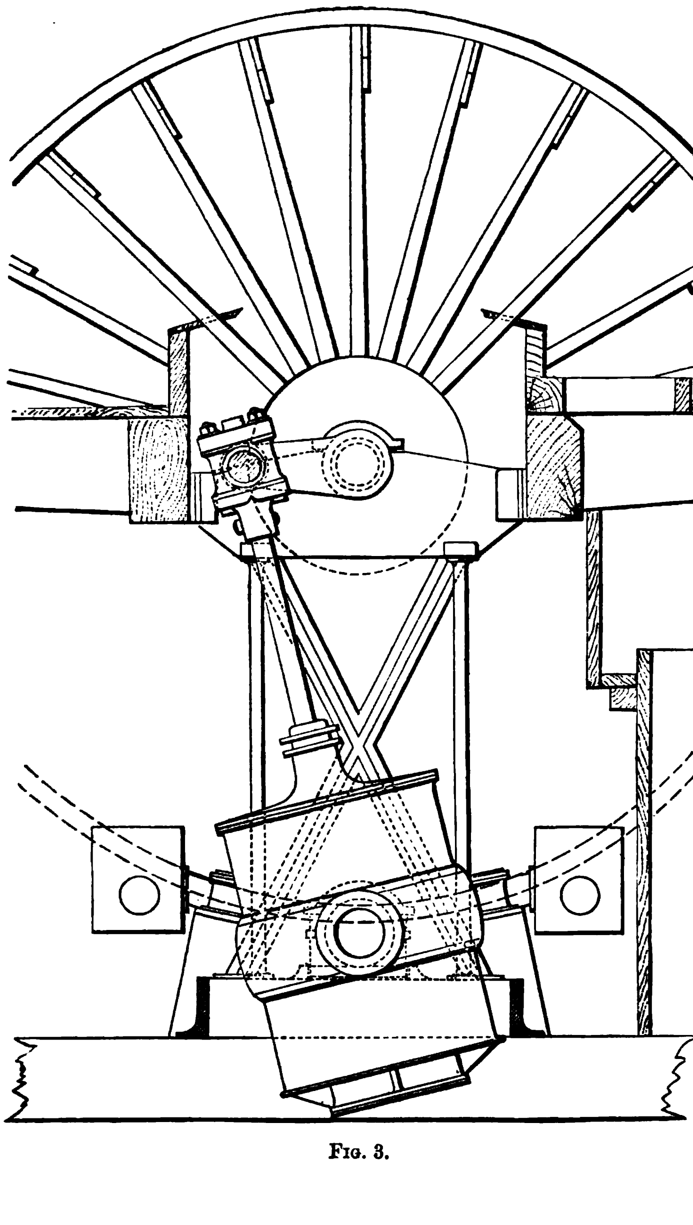

FIG. 3.

Maudslay's double-cylinder engine.—One of the first of the direct-acting engines was Maudslay's double-cylinder engine, a section of which is shown in Fig. 2. It consisted of two equal cylinders side by side, the piston-rods from the two cylinders being connected to a single crosshead. In order to get sufficient length of connecting-rod, the crosshead was of peculiar form and passed down between the cylinders, having a journal at its lower end, on which one end of the connecting-rod worked, the other end being attached to the crankpin.

Oscillating engine.—The simplest and most compact type of engine for driving paddle-wheels is that known as the oscillating engine. This type of engine was first adapted for marine purposes by Messrs. Maudslay, Sons & Field, who, in 1828, fitted a pair of oscillating engines into a steamship called the 'Endeavour,' which was the first ship so fitted. Several other ships were also fitted with oscillating engines by Messrs. Maudslay, but the advantages of this type did not appear at the time to be appreciated. Some years afterwards it was however adopted and perfected by the late eminent engineer, Mr. John Penn, with whose name it is now generally associated. Its general arrangement is shown in Fig. 3. In these engines the connecting-rod is altogether dispensed with, the upper end of the piston-rod being fitted with brasses to work directly on the crankpin, and the cylinder itself is carried on trunnion bearings, to allow the necessary oscillation to suit the motion of the crank. The trunnions are hollow, and the steam is admitted to and exhausted from the cylinders through them. In this type of engine, space and weight have been economised as far as is possible for paddle-wheel engines, and the majority of engines now made for paddle-wheel vessels are on this plan.

The 'Magicienne' was one of the best specimens of

the steam war-vessels of that period. She was fitted with oscillating engines by Messrs. Penn in 1850. The pressure of steam in the boilers was 14 lbs. per square inch, number of revolutions per minute at full power $20\frac{1}{2}$, giving a speed of piston of 287 ft. per minute, with a maximum I.H.P. of 1,300. The total weight of the machinery was 275 tons, or 4·23 cwts. per I.H.P.

The paddle-wheel, though it was an efficient propeller, possessed many practical disadvantages which interfered with progress beyond a certain point. Its performance was much affected by the variation of draught of the ship during a voyage, as the coal and stores were consumed. For fighting ships paddle-wheels were particularly unsuitable. The wheels themselves were exposed to danger from shot and shell, and the paddle-boxes interfered seriously with the training and working of the guns. The shafting and many parts of the engines were considerably above the water-line, much of it above the upper deck; and it is also clear that the paddle-wheel is not a form of propeller adapted for the application of very high powers.

Screw propeller.—The adoption of the screw propeller in lieu of the paddle-wheel was the most important step taken in the progress of marine engineering, for this rendered all subsequent advances possible. Its principal advantages, as compared with the paddle-wheel, are, that it is comparatively little affected by the rolling, or by the variation of the draught of the ship during a long voyage, and it is equally capable of application to either great or small powers. It permits of the engines being kept below the water-line, which is very important in unarmoured warships. Screw engines can be further protected, if necessary, by being kept below an iron or steel armoured deck, with armour gratings in the necessary engine-room hatchways and openings. All the more recent cruisers are thus fitted. With screw engines the decks are also

kept clear for the guns, and the propeller is submerged and protected from shot.

The substitution of the screw propeller for the paddle-wheel began to grow general about the period 1845–50. The screw propeller had been invented long before, but its practical utility had not been generally recognised, and it was still regarded as being in the experimental stage. The first notable experiments as to the comparative efficiencies of paddle-wheels and screw propellers were made in 1840; the 'Archimedes,' with a screw propeller, beating the Dover and Calais paddle-wheel packet-boat 'Ariel' by six minutes in going, and five minutes in returning, under steam and sail. The 'Archimedes' afterwards beat the paddle-wheel steamers 'Beaver' and 'Swallow,' but was beaten slightly by the 'Widgeon.' These trials, however, were not of much importance, beyond showing that it was possible with the screw propeller to obtain speeds at least equal to those with the paddle-wheel, when the engines were working at about the same power. The Admiralty, in 1843, caused some important experiments to be carried out with the screw ship 'Rattler' and the paddle-wheel vessel 'Alecto,' and, in 1849, with the screw ship 'Niger' and paddle-wheel vessel 'Basilisk.' The results in each case were in favour of the screw propeller, and many valuable conclusions were deduced from the trials.

From that time the use of the screw propeller gradually became more and more general, till at the present day it is almost solely employed for marine propulsion, the paddle-wheel only being applied in special cases. It is not too much to say, that ships of the class now traversing the ocean in all directions, both in the royal and mercantile navies, would not have been possible had not the screw superseded the paddle.

Geared engines.—In order to attain the same speed of

ship the screw propeller had to be driven at a much greater speed than the paddle-wheel, and as it was not possible in the then condition of mechanical engineering to drive the pistons at a sufficiently high speed to enable the engine shaft to be connected directly to the propeller shafting, the earlier engines used for working screw propellers were geared so that the screw-shaft was caused to revolve at a much higher rate of speed than the engine shaft. A large spur wheel, keyed on the crank-shaft of the engine, worked into a pinion on the screw-propeller shafting, so that the speed of the engine shaft could be multiplied on the screw-shaft as many times as might be necessary.

Direct engines.—Before long, however, improvements in workmanship, appliances, and mechanical details so far advanced, that the speeds both of piston and of revolution could be sufficiently increased to allow the crank-shaft to be connected directly to the screw-shafting, and now in many marine engines the piston speeds are as high as from 700 to 800 feet per minute. It is probable that in the future of marine engineering the speeds may be increased even beyond this, in order to attain increased economy, and it may become necessary to again introduce gearing, to reduce the velocity of the screw-shafting, instead of, as in the earlier stages, to increase it.

One of the great obstacles that had to be overcome in connecting the crank-shaft of the engine direct to the screw-shafting, when the engines were in a horizontal position, was the close proximity in which the cylinder was necessarily placed to the centre-line of the ship, which made it difficult to get a connecting-rod of suitable length to work between the cylinder and the crank.

Trunk engines.—Mr. John Penn solved this difficulty by his invention of the trunk engine. In this engine a large hollow trunk, cast on the piston, and working

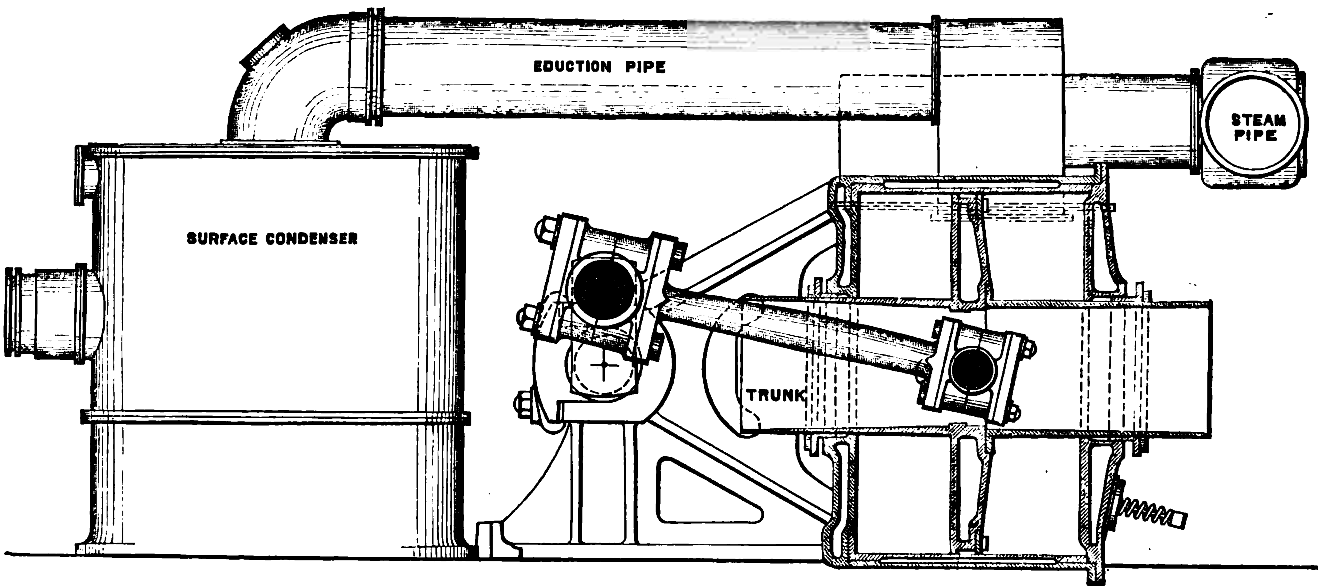

FIG. 4.

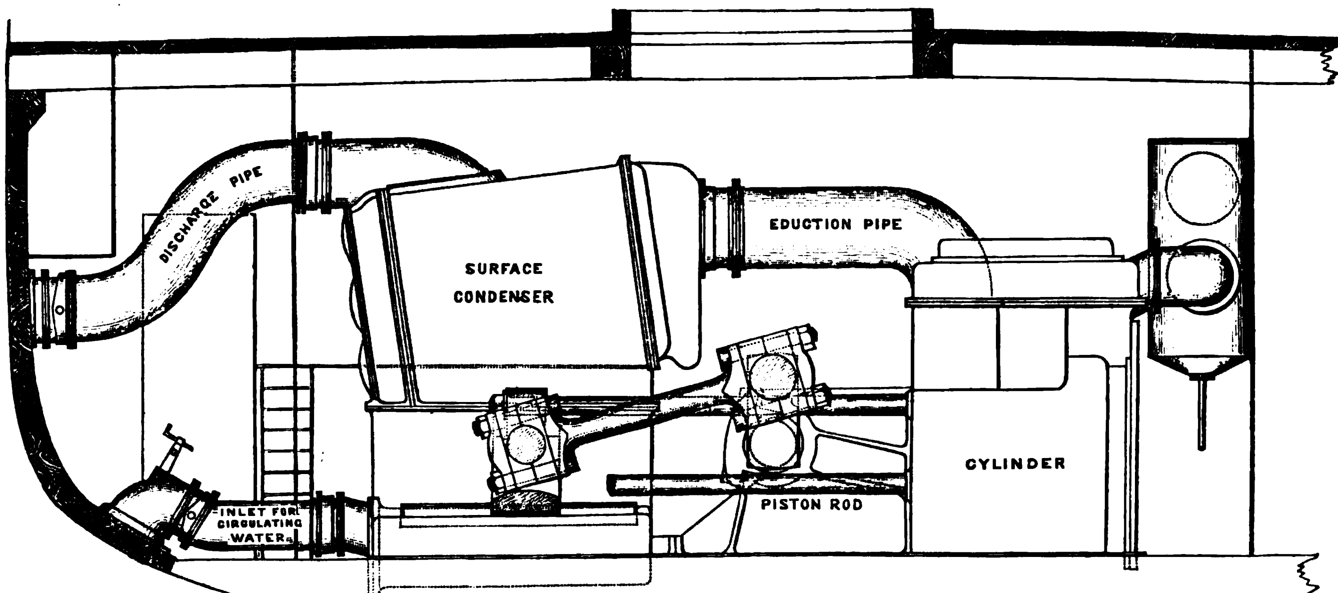
DISCHARGE PIPE
SURFACE CONDENSER
EDUCTION PIPE
CYLINDER
PISTON ROD
INLET FOR CIRCULATING WATER.

FIG. 5.

through a steam-tight stuffing-box on the end of the cylinder, was substituted for the piston-rod, and the connecting-rod was attached directly to a journal or gudgeon in the centre of the piston itself, as shown in Fig. 4. Though the use of a large trunk of this description does not appear at first sight desirable, yet the engines of this type have generally worked in a most satisfactory manner, and they were recognised as amongst the most smooth-working and efficient marine engines employed previous to the introduction of high-pressure steam, for which they do not appear to be so suitable.

Return connecting-rod engines.—This kind of engine was adopted by the majority of marine-engineering firms, to enable the cylinders to be brought close to the crank-shaft when the engines were horizontal. In this arrangement, an illustration of which is given in Fig. 5, there were two rods to each piston, one passing above, the other below, the crank-shaft, to the opposite side of the ship. The further ends of the piston-rods were fixed to a crosshead, having a journal at its centre, from which the connecting-rod worked back to the crank. In the engines of H.M.Ss. 'Monarch' and 'Raleigh,' made by Messrs. Humphrys, Tennant & Co., there are four piston-rods to each cylinder, two passing above and two below the crank-shaft to the crosshead at the opposite side of the ship.

Fig. 6 shows a modified arrangement of return connecting-rod, which has been fitted in some recent ships. This plan has the advantage of a single piston-rod and stuffing-box for each cylinder, and the design of the piston is simplified. The piston-rod is attached to a crosshead, from which two rods are carried, one above, the other below, the shaft, to a similar crosshead on the opposite side, as in the ordinary return connecting-rod arrangement.

The majority of steamers, both war and mercantile, built during the years 1850–60, were fitted with horizontal screw-

propeller engines worked with steam of from 20 to 25 lbs. pressure per square inch. The engines had jet injection condensers, and were not remarkable for economy of fuel, but they were much lighter, and occupied considerably less space, than the paddle-wheel engines that preceded them.

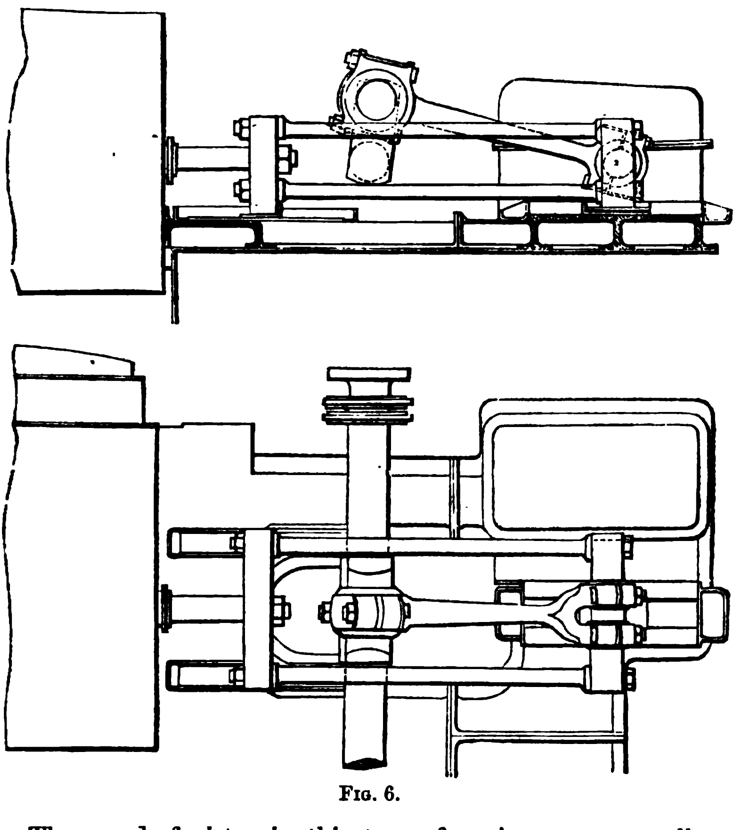

FIG. 6.

The speed of piston in this type of engine was generally about 400 ft. per minute, and the weight of machinery about 3¾ cwts. per I.H.P.

Direct-acting engines.—The direct-acting engine shown in Fig. 7, having the connecting-rod between the cylinder

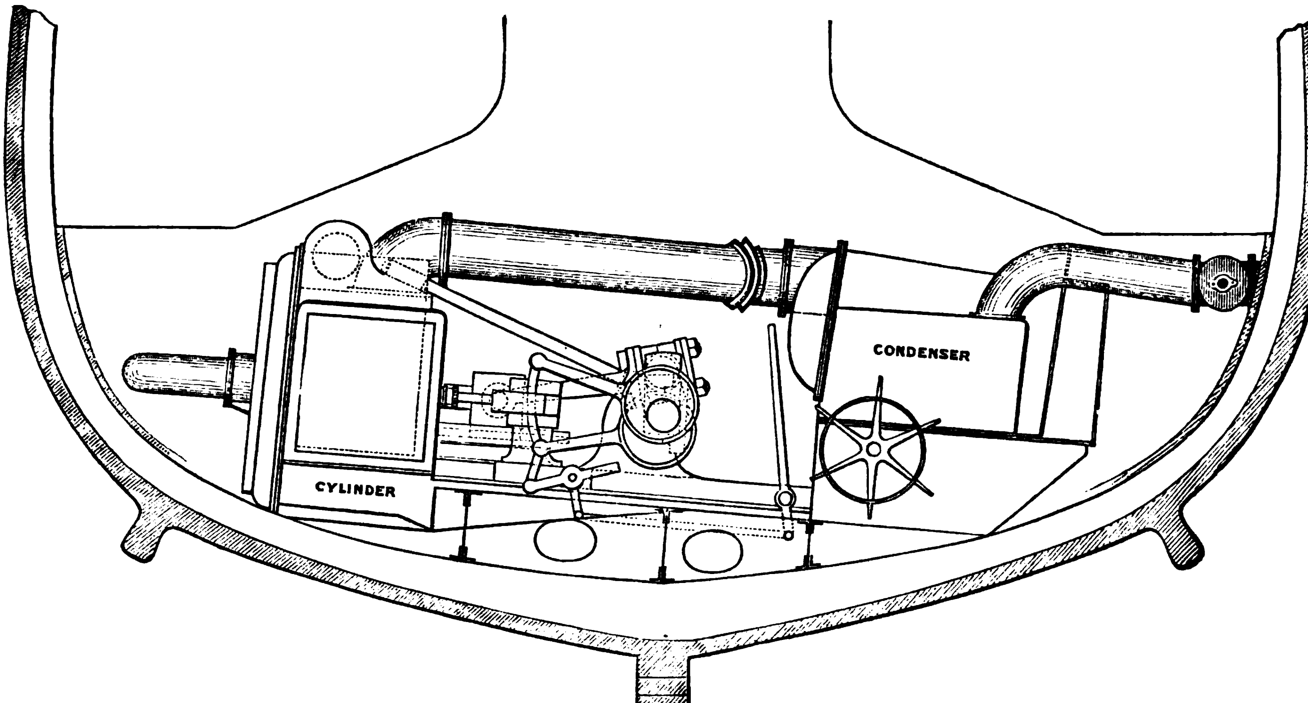

Fig. 7.

and the crank, is the simplest type and the most suitable for general work, and, whenever sufficient room can be obtained, it is usually adopted. Messrs. Humphrys, Tennant, & Co. have almost always employed this plan even for horizontal engines, being satisfied with a shorter connecting-rod than may be obtained with the other types. For vertical engines, with the cylinders at the top working down to the crank-shaft, which are now generally fitted for marine purposes, this type is universally adopted.

Surface condensation.—The adoption of surface condensation, which became general about 1860, formed another important step in marine engineering. Its value consisted not so much in the economy effected by the avoidance of the loss from brining, or blowing out, from the boilers, as on the fact that by its eliminating the element of danger resulting from deposit of solid non-conducting matter on the heating surfaces, it rendered possible the use of high steam-pressures in marine boilers, and thus paved the way for the introduction of the cylindrical boilers and compound engines, now almost universally used. When surface condensation was first introduced, the old flat-sided boilers, made to fit the section of the ship, were still retained. This form of shell is obviously unfit for high steam-pressures; but the boilers were strengthened by fitting additional stays to enable them to carry working steam-pressures of 30 to 35 lbs. per square inch, and the majority of warships built during the years 1860–70 were fitted with surface-condensing engines worked with steam of this pressure. The piston speeds were also considerably increased, especially in the larger ships in which a long stroke could be obtained. With this type of engine the piston speeds varied from 500 to 665 ft. per minute. To promote economy of fuel the cylinders were usually steam-jacketed, and made large enough to allow for considerable expansion at full power, and the boilers were fitted with

superheaters. The average weight of the machinery of this type, including the water in boilers and condensers, was about 3 cwts. per I.H.P.

Compound engines.—We now come to the compound engine, which system has been fitted to nearly all warships since 1870, and which may be regarded as the most general type of marine engine of the present day.

In this type of engine the expansion is conducted in stages carried out in two or more successive cylinders, so that the strains on the framing and journals are decreased and the loss from liquefaction of steam in the cylinders is reduced to a minimum, and the general consensus of opinion and experience points to the following summary of its advantages:—

1. Reduction of the maximum strains on the framing, shafting, and bearings, and consequent reduction of weight and cost.

2. Increased regularity of turning moment, and consequent increased efficiency of the propeller in the water.

3. More economical use of the steam in the cylinders and consequent increase of power from a given expenditure of heat.

The working steam-pressure in the Royal Navy with this type of engine was originally 60 lbs. per square inch. This has been increased from time to time, and the general working pressure in the recently completed ships has been 90 to 100 lbs. In the ships now building the specified working pressure is 130 to 150 lbs. per square inch.

From the adoption of compound engines and higher steam pressures much economy of fuel has resulted. The gain in economy over the ordinary surface-condensing engines worked with steam of 30 lbs. pressure, may be taken to be at least 30 per cent., and the consumption of coal is now scarcely more than one-third that required for the engines in general use prior to 1860. The effect on war-

ships of this great reduction of coal expenditure has been twofold:—

a. The increased distance ships have been able to steam without exhausting their coal supply, has rendered seagoing mastless armour-clad ships possible.

b. The reduced quantity of coal necessary to be carried in the majority of warships has enabled space and weight which would formerly have been required for coals to be devoted to other objects in order to increase their offensive or defensive powers.

The compound engines introduced into the Royal Navy in 1870 were of about 2,000 I.H.P., having two cylinders, one high and one low pressure, side by side. As the size and power of compound engines were increased the dimensions of the low-pressure cylinders became so great that it was found desirable to fit two low-pressure cylinders instead of one, in consequence of the difficulties experienced in obtaining sound castings of large size, and this led to what is known as the *three-cylinder compound engine*, which is simply a modification of the ordinary two-cylinder compound engine.

Vertical engines.—The vertical type of engine, with cylinders at the top and crankshaft below, was adopted for merchant ships before it was introduced into the Royal Navy, because it was a necessity in most warships that all the machinery should be kept below the water-line, and horizontal engines alone satisfied this condition. Vertical engines, however, possess many practical advantages over horizontal engines, especially in connection with the working of the cylinders and pistons. When, therefore, the twin-screw system was adopted for armour-clad ships, vertical compound engines were fitted. By dividing the power into two parts, each set of engines, even in a ship of great power, would be of moderate dimensions, and although the whole of the machinery might not in all cases be entirely

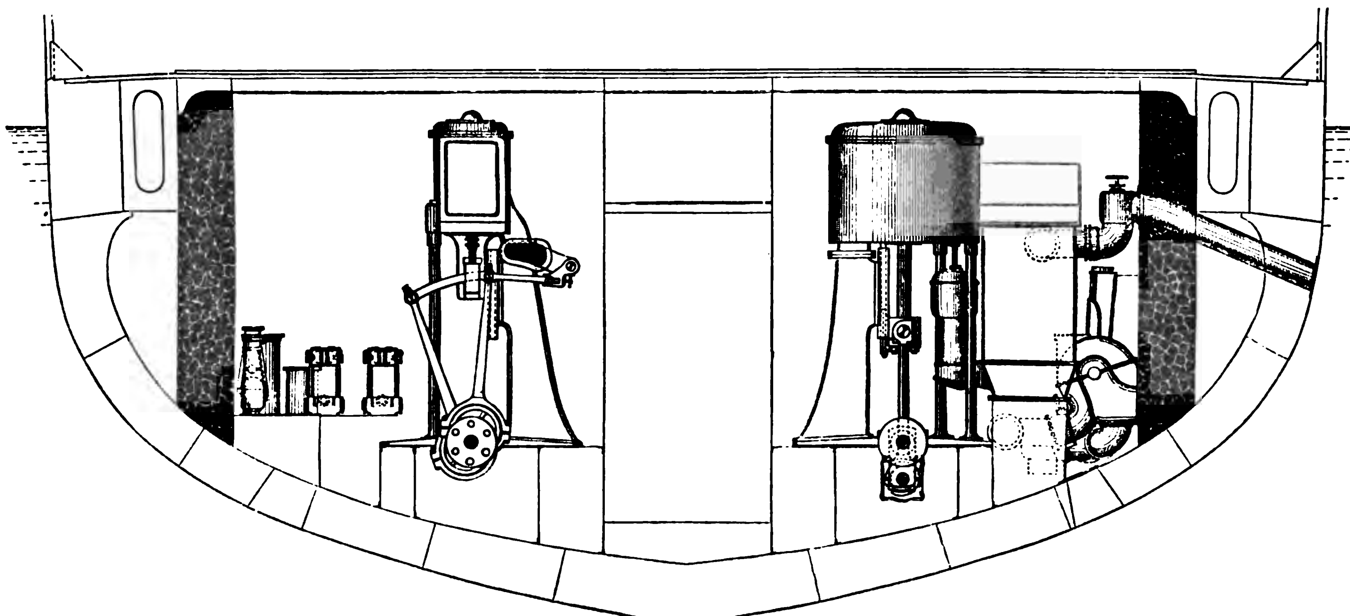

FIG. 8.

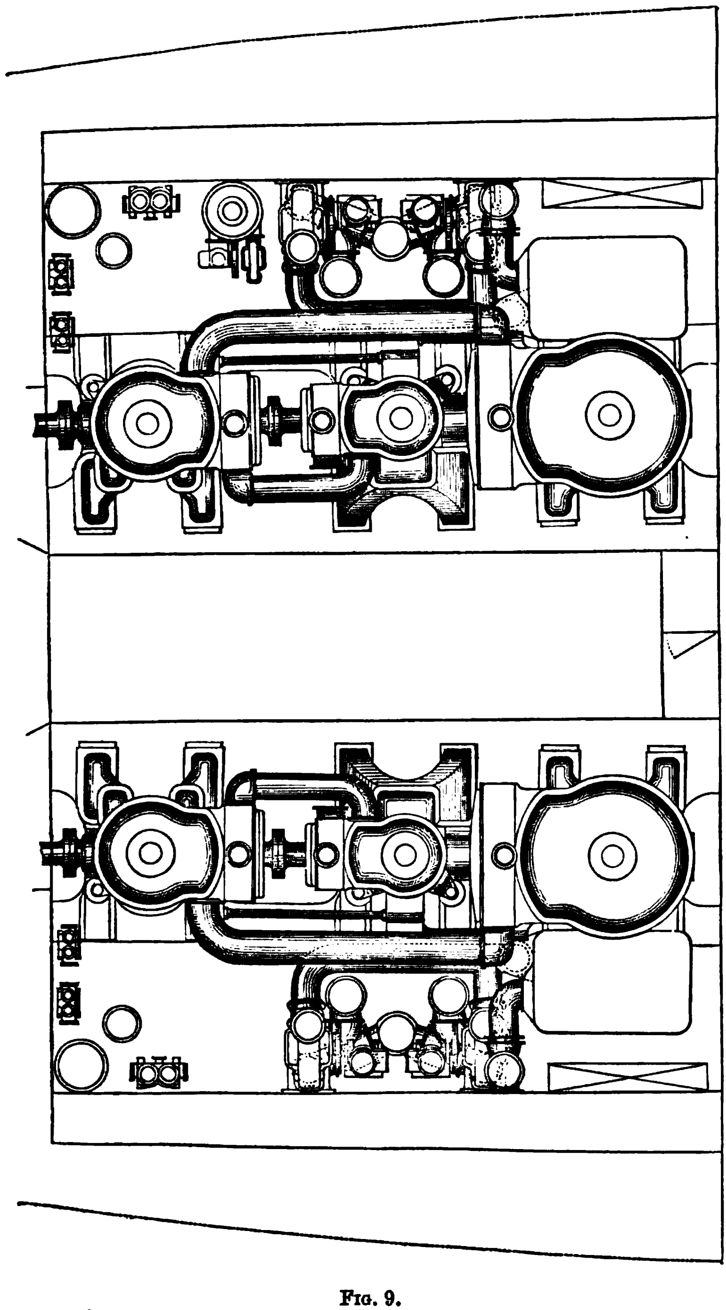

Fig. 9.

below the water-line, the parts above would be protected, not only by armour plating, but by a body of coal in addition, the coal-bunkers being continued on each side of the engine room. This arrangement is shown in Figs. 8, 9, and 10.

Triple expansion engines.—With initial steam pressures above 90 or 100 lbs. per square inch, the variation of temperature in each cylinder of an ordinary compound

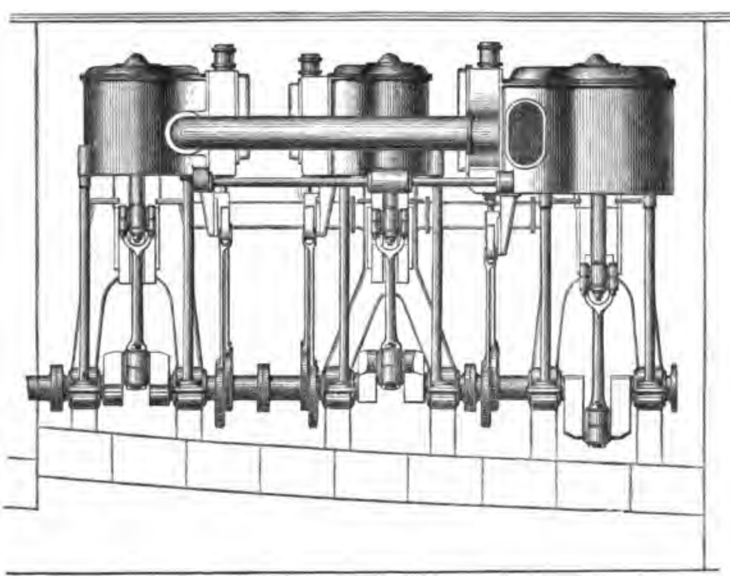

FIG. 10.

engine is so great that the full economy due to the high pressure cannot be attained in consequence of the loss from liquefaction. It has therefore been found desirable to extend the compound system, and divide the expansion into three stages, carried out in separate cylinders, so as to reduce the range of temperature in each cylinder.

Engines on this system are usually known as *triple expansion engines.* They were first introduced by Mr.

A. C. Kirk, senior partner in the firm of Messrs. R. Napier and Sons, Glasgow, who, in 1874, fitted them on board the SS. 'Propontis,' to utilise the steam of 150 lbs. pressure, supplied by Rowan and Horton's patent high-pressure water-tube boilers. These engines gave good economical results, but the boilers unfortunately gave trouble, and were ultimately taken out. Very little further was done in this direction, until, in 1881, Mr. Kirk fitted a set of triple expansion engines on board the SS. 'Aberdeen,' built for Messrs. George Thompson and Sons, of Aberdeen, for their trade to Australia and China. The results in this instance were so satisfactory that other engines of the same type followed, and during the past two or three years the system has been largely adopted in the mercantile marine. The new ships now building for the Royal Navy are being fitted with triple expansion engines. Figs. 8, 9, and 10 show the arrangement of triple expansion engines of 12,000 I.H.P. now being built by Messrs. Humphrys, Tennant, & Co. for H.M. armour-clad turret ships 'Renown' and 'Sanspareil.' The gain in economy effected by the triple expansion engine, worked with steam of 130 lbs. pressure, over the ordinary compound engine worked at 90 to 100 lbs. pressure, may be taken at from 15 to 20 per cent.

In some few cases, in which steam of 150 to 180 lbs. has been used, *quadruple expansion engines* have been fitted, dividing the expansion into four stages, but there is not sufficient evidence yet to show that the additional complication thus introduced is compensated for by any marked gain in economy.

Forced draught.—The conditions of service of ships in the Royal Navy render it necessary to provide for the development of high power and speed on special occasions, such as the events of action, chasing, &c., although the greater portion of the work of the ship has to be performed

at comparatively low powers. It is therefore desirable in warships to provide special means of forcing the boilers when the full speed is required. Formerly the steam jet was used for this purpose, but during the past two or three years the system of forming the stokeholds into closed compartments, and keeping them under air pressure by means of blowing fans, has been adopted with satisfactory results. With the ordinary compound engine and marine cylindrical boiler, the average performance with the steam jet was the development of about 9 to 10 I.H.P. per square foot of firegrate. With closed stokeholds worked under very moderate air-pressures from 16 to 17 I.H.P. per square foot of firegrate can be obtained, thus increasing the steam-generating powers of the boilers by 60 to 70 per cent. Details of the fittings required for this purpose are given in Chapter V. The introduction of forced draught has enabled the weight of machinery to be considerably reduced, and the average weight of machinery for modern warships is not more than 2 cwts. per I.H.P. developed with forced draught.

From this brief sketch a general idea may be formed of the progress that has been made in marine engineering during the past fifty years. The machinery of the 'Rhadamanthus,' built in 1832, weighed 275 tons and developed 400 I.H.P. In modern warships, machinery of the same weight would, under forced draught, be capable of developing at least 3,000 I.H.P., and the space occupied would be considerably less. Another important feature in the progress made is the great increase in the total engine-power now available for the propulsion of vessels. For example, in H.M.S. 'Terrible,' which, in 1845, represented the finest type of steam warship of the day, the maximum I.H.P. was less than 2,000 and her speed about 10 knots. In the despatch-vessel 'Iris' two sets of engines are fitted, together capable of developing 7,700 I.H.P. and of driving the ship at a

speed of $18\frac{1}{2}$ knots per hour. In the twin-screw armoured ship 'Rodney,' of 10,000 tons displacement, the average I.H.P. developed during a four hours' full power trial under forced draught, made on June 13, 1885, was 11,200, and in the armour-clad ships 'Renown' and 'Sanspareil' now building, the guaranteed minimum indicated horse-power is 12,000.

CHAPTER II.

WORK AND EFFICIENCY.

Force.—Force is that which acts in producing or resisting motion in a body; and may be represented by a pressure, or a pull, and expressed generally as being equivalent to so many pounds weight.

Work.—A force is said to perform work when by its action resistance is overcome and motion produced. This union of pressure and motion is essential to the conception of *work*. However great the pressure applied, unless the body acted on be moved no work is done. Work is measured and expressed as a definite quantity by multiplying the amount of the resistance overcome—or, in other words, the force opposing the motion—by the distance through which the force acts. In British units, the distance is usually measured in feet and the force in pounds avoirdupois; and quantities of work, found by multiplying force by distance, are generally said to consist of so many foot-pounds, which are the units of work.

Foot-pound.—The term 'foot-pound,' which is taken to represent the unit of work, is a very convenient one, implying the combination of force and motion which is the essential condition for the performance of work. A foot-pound represents the raising of a weight of one pound one foot high, or more generally the exertion of a pressure of

one pound through the distance of one foot. If 20 pounds be raised 50 ft., the amount of work performed is represented by $20 \times 50 = 1,000$ foot-pounds.

Sometimes for convenience other units of work are used, but they are all formed on the same basis and expressed in a similar manner. For example, the work performed in raising one ton one inch is sometimes called one inch-ton, and it is equal to 2,240 inch-pounds or $\frac{2240}{12}$ foot-pounds. The work of lifting one ton one foot is one foot-ton, and so on. It will be seen that the different terms used are self-explanatory and are convertible one to another. The foot-pound is, however, the general unit, the others only being employed for convenience in special cases.

Efficiency.—In every machine there are always certain causes acting that produce waste of work, so that the whole work done by the machine is not usefully employed, some of it being exerted in overcoming the friction of the mechanism and some wasted in various other ways. The fraction representing the ratio that the useful work done bears to the total energy expended by the machine is called the *Efficiency of the machine*; or

$$\text{Efficiency} = \frac{\text{Useful work done}}{\text{Total energy expended}}.$$

In the marine steam-engine, in which the useful work is measured by its propelling effect on the ship, there are four successive stages, in each of which a portion of the initial energy is wasted, and these four causes all tend to decrease the efficiency of the engine as a whole.

In the first place, only a portion of the heat yielded by the combustion of the coal in the furnaces is communicated to the water in the boiler, the remainder being wasted in various ways. The fraction of the total heat evolved by the combustion of the coal, that is transmitted to the water in the boiler, is in ordinary cases not more than

from $\frac{5}{10}$ to $\frac{1}{10}$. This fraction is called the *Efficiency of the boiler*.

Secondly. The steam, after leaving the boiler, has to perform mechanical work on the piston of the engine; but this work, in consequence of the narrow limits of temperature between which the engine is worked, is only a small fraction of the total heat contained in the steam—say from $\frac{1}{5}$ to $\frac{1}{20}$, according to the kind of engine and rate of expansion employed. This fraction, representing the ratio of the mechanical work done by the steam to the total amount of heat expended on it, is called the *Efficiency of the steam*.

Thirdly. In the engine itself, a part of the work actually performed by the steam on the pistons is wasted in overcoming the friction of the working parts of the machinery and in working the pumps, etc. The remainder is turned into useful work in driving the propeller. The fraction, representing the ratio that this useful work bears to the total energy exerted by the pistons, is called the *Efficiency of the mechanism*.

Fourthly. The propeller, in addition to driving the ship ahead, expends some of the energy transmitted to it in agitating and churning the water in which it acts, and the work thus performed is wasted; the only useful work being that employed in overcoming the resistance of the ship and driving her ahead. The ratio of this useful work to the total energy expended by the propeller is called the *Efficiency of the propeller*.

The resultant *Efficiency of the marine steam-engine* is made up of the four efficiencies just stated, and is given by the product of the four factors representing respectively the efficiencies of the boiler, the steam, the mechanism, and the propeller. Any improvement in the efficiency of the marine steam-engine, and, consequently, in the economy of

its performance, is therefore due to an increase in one or more of these elements, and we propose to treat the several points as far as possible in the order named and in each case to describe the efforts that have been made to increase the efficiency.

CHAPTER III.

NATURE AND PROPERTIES OF HEAT.

In order to comprehend the principles on which the construction and performance of the steam-engine depend, and the object of the various improvements that have from time to time been introduced, it is necessary that the true nature and properties of heat should be known. We will therefore, as concisely as possible, state the principal points relative to this subject that are necessary to be borne in mind, in order that the succeeding chapters may be clearly understood.

Temperature.—The temperature or hotness of a body may be defined as the extent to which it may be capable of communicating sensible heat, or heat that may be felt, to other adjacent bodies.

When two bodies of different temperatures are placed in contact with each other, it is a well-known fact that the hotter body becomes cooler and the colder body hotter, till at length the two bodies become of the same temperature, after which no change in the temperature takes place. This is caused by the passage of heat from the hotter to the colder body, and shows clearly that heat is something that can be transferred from one body to another, so as to diminish the amount of heat in the former body and increase it in the latter.

When heat is added to or abstracted from a body, one

of the two following effects is produced: either the temperature of the body is altered or its state is changed. For example, if heat be added to water under the atmospheric pressure, the temperature is increased until it reaches 212° Fahr. After this the addition of heat does not further increase the temperature, but after a time causes the water to evaporate and become steam—that is, it changes the condition from that of a liquid to that of a gas. Again, if heat be abstracted from water, the temperature is reduced till it reaches 32° Fahr., after which the diminution of heat does not further decrease the temperature, but changes the condition of the water from the liquid to the solid state, forming ice. The quantities of heat passing from one body to another can thus be estimated by the effects produced, so that it is clear that heat is something that can be both transferred and measured.

These properties, however, do not constitute it a substance. The true nature of heat has been determined by experiments on friction. It is a matter of common observation that the work expended in friction is apparently lost—that is, it appears no longer in the form of mechanical work; but at every place where friction occurs, heat is developed, and the greater the friction the greater is the amount of heat produced. Experiments have shown that the amount of heat generated by friction is exactly equivalent to the amount of work lost, and it is therefore clear that heat is of the same nature as mechanical work—that is, it is one of the forms of energy.

The theory that heat and mechanical work are identical in essence, though different in form, may be apprehended from the well-known facts that heat, by expanding bodies, is a source of mechanical energy; and, conversely, that mechanical energy when expended either in compressing bodies or in friction is a source of heat.

Thermal unit.—The unit by which heat is measured

is called a thermal unit, and in British measurements represents the quantity of heat necessary to raise one pound of water at its maximum density, which corresponds to a temperature of about 39° Fahr., through one degree Fahr.

Joule's equivalent.—The honour of determining the exact relation between heat and mechanical work belongs to Mr. Joule, who proved, by an elaborate series of careful experiments on the friction of oil, water, mercury, and other substances, that one thermal unit is equal to 772 foot-pounds of mechanical work—that is, that the quantity of heat necessary to raise the temperature of one pound of water at its maximum density, one degree Fahr., can be made to perform work equal to the raising of 772 lbs. one foot high. In honour of the discoverer this important constant, 772, expressing the relation between heat and mechanical work, is called Joule's equivalent, and is frequently denoted by the letter J.

Thermo-dynamics.—The science of thermo-dynamics consists in the reduction of the laws connecting heat and mechanical work to a physical theory, and it is on this science that all investigations of the principles and action of the steam-engine depend. The two fundamental laws of thermo-dynamics are—

1. Heat and mechanical energy are mutually convertible, and heat requires for its production, and produces by its disappearance, mechanical energy in the proportion of 772 foot-pounds for each unit of heat.

2. If the total actual heat of a homogeneous and uniformly hot substance be conceived to be divided into any number of equal parts, the effects of those parts in causing mechanical work to be performed are equal.

The second law of thermo-dynamics may be thus expressed as regards absolute temperatures :—

If the absolute temperature of any uniformly hot

substance be divided into any number of equal parts, the effects of those parts in causing work to be performed are equal.

By absolute temperature is meant the temperature measured from the absolute zero of the air thermometer, which is 461° below the ordinary zero in Fahrenheit's scale.

Communication of heat.—Heat may be communicated from one body to another in three different ways, viz., radiation, conduction, and convection.

Radiation.—Radiant heat is given off from hot bodies in straight lines, and the rays of heat are subject to the same laws as the rays of light.

As far as the generation of steam is concerned the useful radiation is confined to the furnace, the crowns and sides of which, intercepting the rays of heat from the burning fuel, become themselves heated, and the heat passes through them to the water in the boiler. The amount of heat given off by radiation from burning coal is estimated to be about one-half the total heat produced by the combustion of the coal. It is very important, therefore, to intercept the radiant heat from the coal, to insure that as far as possible the whole of the heat diffused in this way should be transmitted either directly or indirectly to the water in the boiler, and not wasted on the external air or other bodies.

Radiation is an important item to be considered with reference to the economical employment of steam, for it always causes a certain loss of heat, and unless proper precautions are taken this loss may become very considerable.

The surfaces of the boilers, steam-pipes, cylinders, &c., when the engines are at work, are very much hotter than the external air, and consequently, in order that loss of heat by radiation may be avoided as far as possible, all

those surfaces should be clothed with some non-conducting material. Hair-felt has been largely employed for this purpose, and this is usually kept in its place by an outer covering of canvas, wood, or sheet-iron. Preparations of cork and other non-conducting materials have also been used. These substances, however, when applied to very hot surfaces are in danger of being burnt away, and various incombustible non-conductors, such as silicate cotton, fossil meal, carbon cement, &c., have recently been used with success.

The efficient clothing of the hot surfaces is of great importance, and if it be neglected the economical working of the machinery may be seriously impaired.

Conduction.—The second way in which heat may be transferred from one body to another is by conduction. There are two kinds of conduction, called respectively internal and external conduction. The conduction that takes place between the contiguous particles of one continuous body is called internal conduction. The term external conduction is used when heat passes through the points of contact of two distinct bodies.

Rankine states that in boiler plates and flues the resistance offered to heat entering and leaving the surfaces of the plates is in general so much greater than the resistance offered to its passage through the body of the plate, that the nature and thickness of the plate have no appreciable effect on the rate of conduction through it, so that the rate depends on the difference of temperature of the fluids on the two sides of the plate. This, however, does not appear to agree entirely with general observation, thick furnace-plates apparently receiving a slightly thicker coating of scale than thin ones, when worked under similar conditions; which would tend to show that the ebullition was less on the thicker plates, and consequently that their evaporative power was inferior to that of the

thin plates. The difference, if any, is probably not very great, and for all practical purposes the following approximate rule given by Rankine may be safely taken for the rate of conduction through boiler plates and flues :—

$$q = \frac{(T_1 - T_2)^2}{a},$$

where q = rate of conduction through the plate, in thermal units per square foot of surface per hour;

T_1 and T_2 = the temperatures on the opposite sides of the plate;

and a = a constant, which is in ordinary cases between 160 and 200.

Convection.—The third method of transfer of heat is by convection. This is the way in which gases and liquids are heated. Conduction, in the true sense of the word, is very slow in liquids, and almost, if not wholly, inappreciable in gases. When heat is applied to the bottom of a vessel containing a fluid, the particles in contact with the bottom are first heated, and rise through the superincumbent mass of fluid, allowing cooler particles to take their place, which become themselves heated, to rise and circulate through the mass in a similar manner.

It is essential that circulation and mixture of all the particles of a fluid mass should take place to cause the temperature to be uniform throughout the mass. In order that heat may be efficiently transmitted through boiler plates and flues, each of the fluids in contact with them—viz. the water on the one side and the heated gases on the other—should have free circulation, so that the particles in contact with the plates should not be considerably different in temperature from those at some distance from the plates. Boilers are, therefore, often fitted with circulating plates to set up currents in the water, and with

bafflers and bridges in the flues to break up the currents of hot gas and form eddies, in order to promote circulation and mixture in the respective fluids.

In order to render the transfer of heat from one fluid to another through a plate most efficient the general motion of the two fluids should be in opposite directions to each other, so that the hottest parts of the two fluids are opposed to each other and the minimum difference of temperature is the greatest possible. Consequently, in a boiler, in order that the best results may be obtained, the feed-pipes and circulating plates should be so arranged that the general motion given to the water should be as far as possible in the opposite direction to that of the hot gases on their way to the funnel. For the same reason, the action of a surface condenser is most efficient when the cold water, for condensing the steam, enters the condenser at the end at which the condensed steam leaves it, so that the water opposed to the entering steam will have been heated to some extent by its passage through the condenser.

CHAPTER IV.

APPLICATION OF HEAT TO WATER.

We will now consider the effects produced by the application of heat to water. At first the temperature of the water is raised. The particles of water in contact with the plates or flues through which the heat is transmitted become heated, and rise and circulate through the mass of water, their places being taken by cooler particles, till at length the whole of the water is raised to the boiling point, by the convection of heat.

Sensible heat.—The heat added to the water up to the boiling point is generally called sensible heat, its effect being simply to change the temperature and not the state of the water, and its amount may be measured by the thermometer.

Latent heat.—After this, in order to convert the boiling water into steam, a large quantity of heat has to be expended (approximately between five and six times the quantity necessary to raise the temperature of the water from the freezing to the boiling point), which does not produce any increase in the temperature. This heat is known as latent heat.

During the period when the material theory of heat was held—heat being considered as a kind of substance called caloric—it was supposed that the quantity of heat required for evaporation became hidden, or latent, in some

way, during the change from the liquid to the gaseous state, and that it again became sensible or tangible on the reverse process being performed. We now, however, know that heat is not matter, but simply the result of violent mechanical action due to the motion of the molecules of the body; and the development of the science of thermodynamics has shown that this amount of heat, instead of being lost or hidden, is simply transformed into mechanical work, which is performed in changing the state of the water from that of a liquid into that of a gas.

The term latent heat has been retained for the sake of convenience, but it must be understood as an expression that means neither more nor less than *the quantity of heat that must be expended on a body in a given state to change it into another state without altering its temperature.*

Boiling point.—The boiling point, or the temperature of ebullition, of any liquid may be defined as that stage in the addition of heat to the liquid at which the pressure throughout the mass of liquid is not greater than the pressure of saturation due to the temperature, so that the communication of heat and the removal of vapour are continuous.

The temperature of the boiling point depends on the *pressure* under which the liquid is evaporated. The greater the pressure the higher is the temperature at which the liquid boils.

The boiling points of fresh water are approximately as follow :—

Under a pressure of	**1**	**atmosphere**	**212° Fahr.**
,,	**2**	,,	**249** ,,
,,	**3**	,,	**273** ,,
,,	**4**	,,	**291** ,,
,,	**5**	,,	**306** ,,
,,	**6**	,,	**319** ,,
,,	**7**	,,	**330** ,,
,,	**8**	,,	**339** ,,
,,	**10**	,,	**357** ,,

and at a pressure of twelve atmospheres, which is about the highest pressure yet used in ordinary marine boilers, the boiling point of fresh water is 371° Fahr.

The boiling point of a liquid is also affected by its *density*. Solid matter dissolved in the liquid, as, for example, salt in water, resists ebullition and increases the temperature at which the liquid boils. Ordinary sea-water contains about $\frac{1}{33}$ part of solid matter, and this raises the temperature of the boiling point by 1·2° Fahr., so that the boiling point of sea-water under the atmospheric pressure, instead of being 212° Fahr., is 213·2° Fahr. The density of the water in marine boilers in ordinary work is rarely allowed to exceed three times that of sea-water, and at this density the boiling point at atmospheric pressure would be 215·6° Fahr.

The term *boiling point* in this book, when no qualification is used, will be taken to mean the boiling point under the atmospheric pressure.

Total heat of evaporation.—The total heat of evaporation is the sum of the latent and sensible heat of evaporation. We will use the term *total heat of evaporation* to denote the quantity of heat necessary to raise one pound of water from the freezing point, 32° Fahr., to a particular temperature, and to evaporate it at that temperature.

The heat required to raise the temperature from the freezing point to the temperature of evaporation is called the sensible heat, whilst the latent heat is the quantity of heat necessary to evaporate the water at the given temperature.

If H represent the total heat of evaporation, L the latent heat, and S the sensible heat, we have

$$H = L + S.$$

The amount of the latent heat diminishes as the temperature of evaporation is increased. The following

table gives the latent heat of one pound of steam at different temperatures of evaporation:—

Under a pressure of	1	atmosphere . .	966 thermal units.
,,	2	,, . .	940 ,,
,,	3	,, . .	923 ,,
,,	4	,, . .	910 ,,
,,	5	,, . .	900 ,,
,,	6	,, . .	891 ,,
,,	7	,, . .	883 ,,
,,	8	,, . .	876 ,,
,,	10	,, . .	862 ,,
,,	12	,, . .	851 ,,

The latent heat of a pound of steam in thermal units at any given temperature of evaporation, T, is given by the following approximate formula:—

$$L = 966 - 0{\cdot}7\ (T - 212)$$

Though the latent heat of steam is diminished as the temperature of evaporation is increased, the increase of temperature, or, in other words, the increase of sensible heat, is greater than the decrease of the latent heat, so that the *total heat* of the steam is gradually increased. For example, the latent heat of evaporation of one pound of steam at the atmospheric pressure is 966 thermal units. The sensible heat measured from the freezing point is $212 - 32 = 180$ thermal units, so that the total heat of evaporation is equal to $966 + 180 = 1146$ thermal units. At a pressure of two atmospheres the latent heat is 940 and the sensible heat 217 thermal units, making the total heat of evaporation equal to $940 + 217 = 1157$ thermal units.

The following table gives approximately, in thermal units, the latent, sensible, and total heat of evaporation of one pound of steam up to a pressure of twelve atmospheres, which represents a steam pressure of about 160 pounds per square inch above the atmosphere.

	L	S	H
At a pressure of 1 atmosphere . .	966	180	1146
" 2 " . .	940	217	1157
" 3 " . .	923	241	1164
" 4 " . .	910	259	1169
" 5 " . .	900	274	1174
" 6 " . .	891	287	1178
" 7 " . .	883	298	1181
" 8 " . .	876	307	1183
" 10 " . .	862	325	1187
" 12 " . .	851	339	1190

We see from this table that notwithstanding the decrease in the latent heat the total heat of evaporation slowly increases. This point will be found to be important in considering the expansive properties of steam, and one that will explain some rather curious phenomena.

The total heat of evaporation at any temperature T, may be calculated from the following formula :—

$$\begin{aligned} H &= L + S \\ &= 966 - 0{\cdot}7\ (T - 212) + (T - 32) \\ &= 1082 + 0{\cdot}3T \end{aligned}$$

This formula shows very clearly the rate of increase in the total heat as the temperature of evaporation is raised.

The different stages in the evaporation of water, just discussed, will perhaps be more clearly understood and summarised in the following manner.

Suppose one pound of water, at a temperature of 32° Fahr., to be contained in a cylinder, open at the top, in which a piston works steam-tight. The pressure on the water, which consists of the weight of the piston in addition to the atmospheric pressure, is represented by P, or say p pounds per square inch on the piston.

In Fig. 11 three stages in the process of evaporation, when heat is applied to the bottom of the cylinder, are shown. A B is the cylinder, and the piston C is kept on

the surface of the water or steam by a total pressure, P, acting in the direction indicated by the arrows.

When heat is applied no sensible movement of the piston occurs until the temperature of evaporation is attained, which temperature will depend on the amount of the pressure p produced by the piston on the water. This part of the process, viz. the raising of the temperature from the freezing to the boiling point, is shown in Stage 1.

From this point the *temperature* remains constant, and steam is given off at the pressure p pounds per square inch, which causes the piston to rise, as shown in

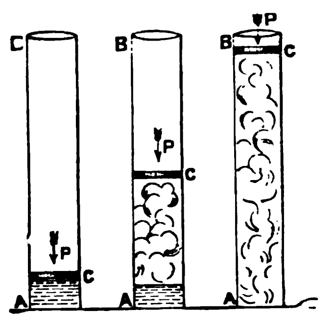

FIG. 11.

Stage 2, the piston continuing to rise during the addition of heat, as more and more of the water is evaporated until the whole of the water is turned into steam, which is the condition supposed in Stage 3.

Stage 1 represents the addition of sensible heat to the water, and Stage 2 the addition of latent heat, the effect of the total heat expended being shown in Stage 3, in which the temperature of the steam will be simply due to the pressure of evaporation p.

If, after all the water is evaporated, more heat be applied to the steam in the cylinder, the pressure P on the

piston remaining the same, the temperature will again begin to increase, and may be raised to any amount if sufficient heat be supplied. The steam in this case is said to be superheated.

In Stage 1, therefore, we have increase of temperature without increase of volume; in Stage 2, increase of volume without increase of temperature; in Stage 3, if further heat be supplied, the pressure being kept constant, we should have increase of both temperature and volume. If in Stage 3 the vessel be closed so that the volume remains constant while additional heat is supplied, the pressure will increase at a rate approximately proportional to the increase in the absolute temperature of the steam.

For a complete theory of the steam-engine a thorough knowledge of all these stages would be required, but the properties of steam in the last stage, viz. when heated without being in contact with the water from which it was generated, has not yet been fully ascertained, and in investigations on the action of superheated steam it is generally assumed that its properties are similar to those of a perfect gas.

The experiments of Regnault and others have, however, given almost all the necessary information with reference to the other stages, which are those that will most concern us in this treatise, as the use of superheated steam for marine engines is very limited, and is becoming entirely discontinued as higher pressures are being introduced, for reasons that will be explained further on.

PART II.

THE BOILER.

CHAPTER V.

COMBUSTION OF COAL AND ECONOMY OF FUEL.

IN this chapter we will consider the processes through which fuel passes during combustion, and the precautions necessary to insure economy.

The heating elements in ordinary fuel are hydrogen and carbon. Sulphur is sometimes present in small quantities, but its heating effect is generally inappreciable.

Combustion.— Combustion is simply a rapid chemical union of oxygen with the hydrogen and carbon in the coal, heat being evolved during the process. There are two distinct stages in the combustion of coal, viz. the formation and combustion of the gases, and the combustion of the solid residue. It is frequently assumed that the gaseous constituents of the coal are given off *during* the combustion. This, however, is not the case. The distillation of the gases is an entirely distinct operation from the combustion.

We will consider the processes of combustion of bituminous coal, which is the most difficult to burn economically in ordinary furnaces on account of the inefficiency of

the ordinary arrangements for consuming the gaseous products.

When a fresh charge of coal is thrown on a bright fire it at first absorbs heat from the fire to liberate the gases. Until all the gas is separated from the solid part of the fuel no combustion takes place, but the coal remains in an unburnt and comparatively cool state, so that, unless proper precautions are taken to consume the gas, it may be a source of loss instead of gain, because heat is abstracted from the fire to effect its liberation. This gas is the same as the coal gas generated in retorts and used for lighting purposes, and consists principally of light and heavy carburetted hydrogen gases. The former, which is also known by the names of fire-damp and marsh gas, is represented by the chemical symbol CH_4. Heavy carburetted hydrogen or olefiant gas contains twice as much carbon as marsh gas, and is denoted by the formula C_2H_4.

On the application of heat and atmospheric air the hydrogen and carbon are separated, and each enters independently into combination with oxygen, forming steam and carbonic acid gas. From the combining equivalents of hydrogen, carbon, and oxygen, we find that each volume of light carburetted hydrogen gas will require two volumes of air for its combustion, and each volume of the olefiant gas three volumes of oxygen. Taking, then, the lower estimate we see that at least two volumes of oxygen will be required for the complete combustion of each volume of the gas formed in the furnace; and as the oxygen in atmospheric air amounts to only one-fifth of its bulk, ten volumes of air will be required for each volume of gas.

One ton of bituminous coal is estimated to produce on an average about 10,000 cubic feet of gas; therefore 100,000 cubic feet of air must actually combine with the gases produced from each ton of coal to effect their com-

plete combustion. To insure thorough mingling of the air with the gas, in order to effect perfect combustion, it is found that double the quantity of air theoretically necessary must be admitted; so that 200,000 cubic feet of air are required for the combustion of the gases alone produced from one ton of bituminous coal. All this air should be admitted without passing through the burning coal on the bars, which would deprive it of a great portion, if not all, of its oxygen, and depreciate or destroy its value for mixing with the gas.

We now pass on to consider the coke or carbon that remains on the bars after the gases have been disposed of. The air for its combustion must pass between the bars and through the fuel. At first the union of oxygen with the carbon is complete, and carbonic acid gas is formed; but this gas as it rises through the fire takes up more carbon, forming carbonic oxide gas, and unless arrangements are made to consume this gas by adding additional oxygen, a large quantity of heat will be wasted, the products of combustion passing off as carbonic oxide, by which less than one-third the heat is produced that would be yielded if the combustion were complete and the products passed off in the form of carbonic acid. It is therefore necessary, especially with thick fires, to admit air above the fuel, for the complete combustion of the carbon as well as of the gases.

Let us now estimate the quantity of air necessary for the combustion of the carbonaceous portions of a ton of coal. Every 6 pounds of carbon requires, in order to form carbonic acid gas, 16 pounds of oxygen. The volume of air necessary to supply this would be about 900 cubic feet. Taking then 80 per cent. of carbon in the coal, we find that about 240,000 cubic feet of air are required theoretically for the combustion of the solid residue of each ton of coal after the gases have been distilled. Doubling

this, as in the case of the gas, to insure perfect mingling, we get 480,000 cubic feet, which, added to the 200,000 cubic feet required for the gas, make a total of 680,000 cubic feet, which enormous volume of air is necessary for the complete combustion of each ton of coal.

These numbers, though clearly illustrating the immense quantity of air that has to be supplied to the furnaces of steam boilers when at work, are perhaps difficult to remember, so we will give the relative weights of coal and of the air required for its combustion which may be more readily borne in mind.

The following formula is given by Rankine for the weight of air theoretically necessary for the complete combustion of each pound of coal:—

$$A = 12C + 36\left(H - \frac{O}{8}\right)$$

Where A = Number of pounds of air required per pound of coal.
C = fraction of carbon contained in the coal.
H = " " hydrogen " " "
O = " " oxygen " " "

In cases where the draught is produced by artificial means, such as a fan or blowing engine, this quantity must be increased by one-half; and in cases of natural or ordinary chimney draught it must be doubled in order to insure perfect mingling.

In practice it is not necessary to calculate with great exactness the quantity of air required for the coal, and it is sufficiently accurate for all practical purposes to take 12 pounds of air as the quantity chemically necessary for the combustion of each pound of coal. With natural draught, therefore, 24 pounds of air must be supplied for each pound of coal to insure perfect admixture. With artificial draught 18 pounds of air would probably be sufficient for

each pound of coal, and this adds greatly to the advantages of forced draught, the temperature of the fire being thereby much less decreased, so that the results from the radiation of heat from the burning fuel are greater.

Having now rapidly glanced at the processes of combustion, we will direct attention to the boilers themselves, and consider the points that should receive special attention, in order that economical results may be attained. It will be seen that the processes of combustion are purely chemical, and the first point to claim our attention is the supply of a sufficient quantity of air for the fires. This is of the first importance. No matter how large and roomy the boiler may be, unless provision be made for the proper supply of air, the coal cannot be burnt nor the full power of the machinery developed. We have seen that the combustion of the coal must be divided into two stages, viz. the combustion of the gases and the combustion of the carbon, and each must be arranged for separately. The area of the fire-grate must be designed for the carbon, and the volume of the combustion-chamber for the gases.

Fire-grate.—The carbon being a solid body requires only a limited space, and since its combustion depends on the amount of air supplied to it, and not on the space it occupies, the area of the fire-grate will depend on the draught employed. In locomotive and other boilers with forced draught the rate of combustion is often as high as from 70 to 80 pounds of coal per square foot of fire-grate per hour, whilst in marine boilers, with only the draught due to the heated gases in the funnel, the rate is generally not more than from 20 to 30 pounds per square foot of fire-grate per hour. It is essential for economy that the length of the fire-grate should be kept within such limits that it may at all times be kept well and uniformly covered with coal. With very long fire-grates there is great danger that the back parts of the grates will

not be properly covered, so that volumes of cold air rush in at the backs of the fires, cooling the tubes and causing a considerable waste of heat. This action causes the disagreeable rumbling and rattling noise that is sometimes heard in boilers when the stoking is inefficient, especially if the fires are being forced.

Combustion-chamber.—With reference to the combustion-chamber we have other conditions to consider. It is here that the gases have to be consumed, and consequently allowance has to be made for their expansion; the space above the fuel should, therefore, be made as large as possible. During the past few years the tendency has been gradually to increase the combustion-chambers of boilers even at the expense of the loss of a certain amount of tube-heating surface, and this has been found to increase the economical performance of the boilers. Not many years ago the combustion-chambers, even of large boilers, were often not more than 15 to 18 inches deep from the backs of the chambers to the tube-plates, whilst now they are rarely less than from 20 to 30 inches deep, and in many cases they are even larger than this.

One method of construction that has the practical effect of increasing the size of the combustion-chamber may, perhaps, be referred to here. In this plan the furnaces are arranged in pairs, each pair having a common combustion-chamber. By firing the furnaces alternately, that is, firing one furnace when the adjacent one is at or about its maximum heat, the gas given off from the coal thrown on the fire has the whole volume of the common combustion-chamber of the two furnaces in which to expand, as well as the heat from the adjacent furnaces to prevent the gases from falling below the temperature of ignition. It should always be borne in mind that no amount of heat can consume the gas unless air be supplied, but, on the other hand, if the gases are not kept

up to a certain temperature, called the temperature of ignition, the oxygen of the air will not chemically unite with them and cause combustion to take place.

Total heat of combustion.—The total amount of heat produced by the complete combustion of one pound of carbon is 14,500 thermal units, and this is sufficient to convert 15 pounds of water at a temperature of 212° Fahr. into steam of the same temperature. If the carbon be only imperfectly burned, so that carbonic oxide instead of carbonic acid is produced, the amount of heat generated is only 4,400 thermal units, which is less than one-third of the heat yielded by complete combustion. The evaporative power of hydrogen is 4·28 times as great as that of carbon, the total heat of combustion of one pound of hydrogen being 62,032 thermal units.

The total heat of combustion of one pound of coal is given by the formula

$$h = 14{,}500 \left\{ C + 4{\cdot}28 \left(H - \frac{O}{8} \right) \right\}$$

where h = total heat of combustion of the coal in thermal units, and C, H, and O are the fractional parts of carbon, hydrogen, and oxygen respectively contained in the coal.

If percentages be used instead of fractions, the formula becomes

$$h = 145 \left\{ C + 4{\cdot}28 \left(H - \frac{O}{8} \right) \right\}$$

Evaporative power.—In order to convert one pound of water at a temperature of 212° Fahr. into steam of the same temperature, 966 thermal units must be added to it; therefore, by dividing the total heat of evaporation by 966, we get the number of pounds of water that each pound of coal is theoretically capable of converting into steam. This is called the evaporative power of the coal, and if it be represented by the letter E, we have

$$
\begin{aligned}
E &= \frac{h}{966} \\
&= \frac{14{,}500}{966}\left\{C + 4{\cdot}28\left(H - \frac{O}{8}\right)\right\} \\
&= 15\left\{C + 4{\cdot}28\left(H - \frac{O}{8}\right)\right\}
\end{aligned}
$$

where C, H, and O are the fractional parts of carbon, hydrogen, and oxygen respectively contained in the coal.

If, for example, we apply this formula to a bituminous coal, containing, say, 80 per cent., by weight, of carbon, 5 per cent. of hydrogen, 5·5 per cent. of oxygen, and 9·5 per cent. of ash, &c. which has no evaporative power, we shall find that the total heat of combustion of one pound of this coal is 14,268 thermal units, and this is theoretically capable of evaporating 14·77 pounds of water. Again, if we take one pound of Welsh steam coal, containing, say, 90 per cent. of carbon, 4 per cent. of hydrogen, and 4 per cent. of oxygen, we find that it is theoretically capable of evaporating 15·75 pounds of water. Therefore, if all the heat produced by the complete combustion of the coal could be utilised in the boiler, about 15 pounds of water should be evaporated for each pound of coal burned. In practice, however, we fall very far short of this. The best Welsh coal burnt in ordinary furnaces under favourable conditions only evaporates from 9 to 10 pounds of water per pound of coal, and in general practice in marine boilers the quantity of water evaporated per pound of coal is much less even than this.

The difference between the *available* evaporative power and its *theoretical* evaporative power is due mainly to the following causes:—

1. Waste of unburnt fuel in the solid state.
2. Waste of unburnt fuel in the smoky and gaseous states.

3. Waste by external radiation and conduction.
4. Waste or loss of heat by the hot gas which escapes by the funnel.

The first cause of loss, viz., the waste of unburnt fuel in the solid state, generally arises from brittleness in the fuel and want of care in stoking; the coal breaking into small pieces and falling between the bars into the ashpits. This loss may be prevented, to a great extent at least, by firing evenly and uniformly, disturbing the fires as little as possible, and taking care to burn all the small coal and cinders that may fall into the ashpits. With careless stoking the loss from this cause may be very considerable, but with careful firing and management it has been found by experiment to vary from nothing up to 2½ per cent.

The waste resulting from the escape of unconsumed gases up the chimney can only be prevented by providing a sufficient supply of air to the furnaces admitted by proper appliances in a suitable place, combined with careful stoking. There can be no doubt that this is one of the most difficult problems to be dealt with in arranging for the economical working of the boilers. It is extremely difficult to obtain perfect combustion of the gases, because the backs of the fires are in such close proximity to the entrances to the tubes, that the gases have to be distilled, mixed with air, and consumed in a short period of time, and in a very confined space, for the combustion of the gases cannot be considered to extend more than a few inches beyond the entrances of the tubes.

Smoke-burning.—During the past few years a great deal has been said and written about smoke-burning, and the arrangements fitted to boilers for the supply of air to the furnaces above the fuel for the gases is frequently called the 'smoke-burning apparatus.' These terms are very misleading; what has to be done is not to *burn* the smoke, but to *prevent its formation* by rendering the combustion

complete; and the apparatus fitted is simply an arrangement to admit a sufficient supply of air for this purpose. It is quite possible that in many cases the true nature of these fittings has been overlooked or misunderstood, and it has been conceived that when the emission of smoke has been prevented the desired end has been attained. This, however, may be a grave error. If the combustion be complete, no smoke will be emitted, because the products of combustion will pass off as carbonic acid gas, which is invisible. But carbonic oxide gas is also invisible, and if the combustion be imperfect and the products pass off in this form, no smoke will be visible, but the heat produced will be less than one-third of that yielded by complete combustion.

The great object to be attained is *economy of fuel* by rendering the combustion perfect, and the *prevention of smoke* will then follow as a natural consequence. Looking at the subject in this light, it is clear that all schemes that do not increase the evaporative power of the coal must be considered unsuccessful even if they entirely prevent the emission of smoke. It is not uncommon to see in reports of trials of many plans for the prevention of smoke a remark to the following effect: 'The emission of smoke was prevented, but it was more difficult to keep steam when the apparatus was in operation.' This shows either that the apparatus was imperfect in design, or that it was not used in the most efficient manner. It appears to be immaterial where the air is admitted, if a complete mixture of the air with the gases takes place before the latter are cooled below the temperature of ignition. It should not be admitted in volumes but in small jets, and this is done by causing the air to pass through perforated plates into the furnaces or combustion-chamber, where it mixes, or is supposed to mix, with the hot gases.

It will be unnecessary to describe the details of the different plans that have been adopted from time to time

for the prevention of smoke: the principle involved in all is the same, and they only differ in arrangement.

Fig. 12 shows a fitting that has been used for this purpose to a considerable extent in the Royal Navy. In this arrangement the air is admitted direct into the combustion-chamber through slides and perforated plates below the fire-bars at the backs of the bridges. It has been successful in most cases in preventing smoke, but frequently at the cost of the evaporative efficiency of the boiler. In the first place, the fire-bars have generally to

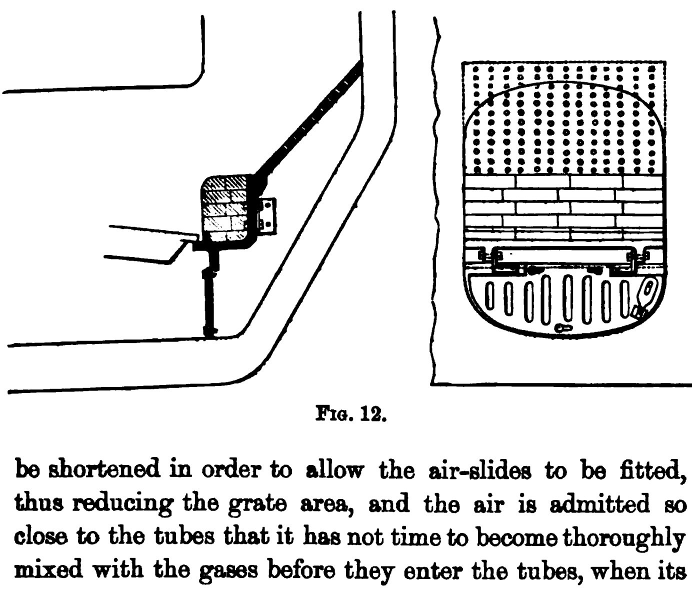

Fig. 12.

be shortened in order to allow the air-slides to be fitted, thus reducing the grate area, and the air is admitted so close to the tubes that it has not time to become thoroughly mixed with the gases before they enter the tubes, when its effect would be to cool them to some extent and thus to further reduce the efficiency of the boiler. This system has consequently become obsolete.

The most efficient plan practically appears to be the fitting of suitable slides in the furnace door or frame, so that the air has to pass over the fuel. This, while allowing more space and time for the mingling of the air with

the gases, has also the advantage of heating the air and preventing its cooling the gases below their igniting temperature, so that there is an increased chance of their complete combustion.

It has been recommended, in order to insure the combustion of the gases, that, in firing, each charge of coal should be laid on the dead-plate at the mouth of the furnace, so that the gases as they were driven off would have to pass over the burning coal at the back, and be thus consumed. After the gases had been all distilled, the coke remaining should be pushed back, the fires levelled, and a fresh charge of coal laid on the dead-plate. This appears plausible, but on examination it will be found to be unsatisfactory and impracticable. It entails much extra work, the furnace doors must be wide open for a considerable time while levelling the fires, allowing volumes of cold air to rush over the fires and through the tubes, and by so much disturbance of the fires considerable waste must ensue from cinders and small coal falling through the bars into the ashpits.

Economy of coal can only be perfected by keeping a continuous supply of coal on the fires and introducing a regular quantity of air for its combustion. The more continuous the supply of fuel the more certainly can it be properly consumed. A continuous supply cannot be kept up while hand stoking is a necessity. The furnace door is wide open for some seconds during firing, allowing cold air to enter, and a large quantity of coal is thrown on the fire, sufficient to last, say, ten to fifteen minutes. The generation of gas is so large and sudden that it is almost impossible to entirely consume it, whereas if the supply of fuel were regular the distillation of the gases would be gradual, and they could be consumed as fast as generated.

It would be a great improvement if mechanical stoking arrangements suitable for marine boilers could be designed,

but the problem is a difficult one. Some plans have been tried in a few merchant steamers which, though not altogether satisfactory, appeared to show good results with respect to economy of fuel. Any arrangement of the kind, before it could be fitted to a ship of war, must be very perfect and reliable; for the engines of these ships have to work at such wide ranges of power, that the apparatus must necessarily be one in which the supply of fuel could be readily increased or decreased to considerable extents, to suit the varying demands of the engines.

In the meantime we are entirely dependent on the care and skill of the stoker, which qualities are frequently difficult to obtain in combination. It is a great mistake to suppose that any man strong enough to throw a shovelful of coal on a fire is fit to perform the duties of a fireman. The United States Government made this mistake some years since, and gave instructions that when a ship was under steam a certain number of men from the deck were to be told off for stokehold duty. After trial of this plan, however, it was found to be most wasteful and inefficient, and in the Report of the Bureau of Engineering for 1873 it was recommended that for the future a class of firemen should be entered to be employed solely on the duties of the engineer department. In point of fact, stokers are skilled labourers, and should be regarded and treated as such, and any extra care and cost involved in securing efficiency in the firemen is soon more than repaid by the increased efficiency and economy of working of the boilers.

The third cause of waste of heat, viz. by radiation and conduction, should in well-constructed furnaces be practically inappreciable. The furnace frame is sometimes made double, and the furnace door should be specially arranged to prevent waste of heat. Fig. 13 shows the construction of a furnace frame and door. The current of cold air entering the frame through the holes in the top outside

passes down the frame into the ashpit, and keeps the outer plate of the frame cool. The two inner plates on the furnace door are perforated to allow air to pass through to the fires, the holes in the two plates not being opposite to each other; so that the heat radiated from the fires through the holes of the inner plate are intercepted by the second plate, and the greater part of the heat thus received is carried back to the furnace by the entering current of air.

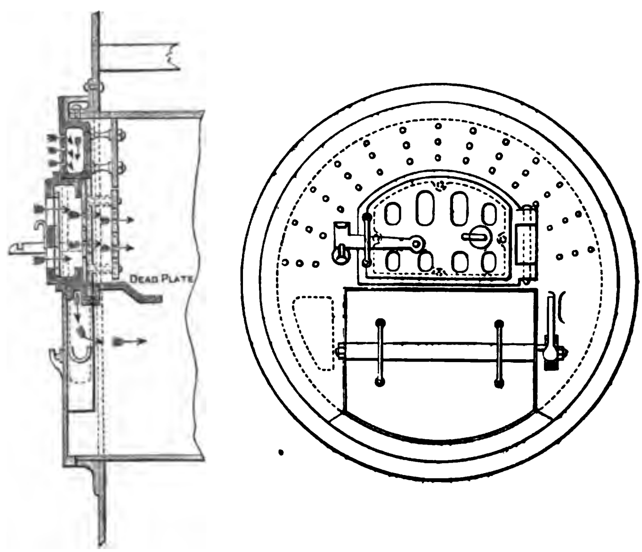

FIG. 13.

The last cause of waste of heat that remains to be considered is that due to the hot gases passing up the chimney. In ordinary marine boilers the draught is produced entirely by the difference in weight between the hot gases in the funnel and that of an equal column of the external air. The weight of the gases discharged from a given chimney per second is a maximum when their absolute temperature—that is, their temperature measured

from the absolute zero of the gas thermometer (which is about 461° below the ordinary zero in Fahrenheit's scale) —is to that of the external air as 25 to 12.

Let T_1 = temperature of the air.
and T_2 = " " gases in the funnel.

Then the weight of the gases discharged from the funnel per second is a maximum when

$$\frac{T_2+461}{T_1+461}=\frac{25}{12}.$$

At the ordinary temperatures this formula would give about 600° Fahr. as the temperature of chimney that gives the most efficient chimney draught. If the temperature be increased beyond this amount, although the velocity of the gas in the chimney would increase, yet its volume would increase in a greater ratio, and consequently the weight of the gases discharged from the chimney, or, in other words, the draught, would be decreased.

The temperature of the fire in a furnace with a draught produced only by a chimney, and supplied with a sufficient quantity of air, is about 2,400° Fahr. above that of the external air; and as the temperature of the gas in the chimney to produce the maximum draught is about 600° Fahr., it would appear that it is never necessary to expend more than one-fourth of the total heat of combustion for the purpose of creating a draught by the chimney. By making the chimney of large enough dimensions as compared with the grate, much less expenditure of heat than this may be made to produce sufficient draught for the rate of combustion in the furnaces.

In some cases a portion of this waste heat is utilised by fitting steam superheaters or feed-water heaters in the uptakes leading to the funnel. These fittings usually consist of a box containing tubes, through or around

which the steam or the feed-water, as the case may be, is conveyed, and absorbs heat from the heated gases on the opposite sides of the tubes, the heat thus added to the steam or feed-water being turned into useful work in the engines.

The several causes of waste enumerated are always at work to a greater or less degree, so that the total quantity of heat that should be yielded by the complete combustion of the coal is not available for transmission to the water in the boiler. The total amount of waste depends greatly on the skill exercised in the design of the boiler and the care displayed in its management. In the best designed tubular marine boiler, with careful manipulation of the fires, the waste usually amounts to nearly one-half the total heat of combustion of the coal. If the management be careless and unskilful the loss may very considerably exceed one-half. This shows that there is a large field in which ingenuity may be exercised, both in improved designs and in more efficient methods of working, in order to reduce each of these causes to a minimum, and thus to increase the economical working of the boilers. The ratio borne by the heat actually transmitted to the water in the boiler to the total heat developed by the combustion of the fuel is called the *efficiency of the boiler.*

Artificial draught.—The question of weight and space occupied by the machinery on board ship is one of great importance. With natural draught only, the rate of combustion per square foot of fire-grate in marine boilers is comparatively slow, even under the most favourable circumstances, as the height of the funnel is necessarily limited, and if the natural draught were alone depended on, the boilers would require much more space than could reasonably be allotted to them, especially in war ships. It is therefore necessary, as explained in Chap. I., that artificial methods should be adopted to increase the rate of

combustion in the fires, and consequently the generative powers of the boilers, in order to obtain the full power of the machinery.

Steam blast.—The most common plan adopted for forcing the draught in marine boilers is by admitting a jet of steam from the boilers at the base of the funnel; this is usually known as the steam blast. It can be very readily applied, and the rate of combustion can thus be materially increased; but it is a very extravagant way of obtaining power, and is injurious both to the boilers and funnels if its use be long continued. It should only be applied when high power is required for a comparatively short period, and when speed is of more importance than economy. The general percentage of increase of power obtained by the steam jet of the proportions usually adopted may be taken to be from 10 to 15 per cent.

The steam blast is especially wasteful and injurious in the case of surface condensation, as the whole of the steam used for the blast is necessarily lost and cannot be returned to the boiler in the form of feed-water, the deficiency having to be made up, generally, by the addition of cold sea-water. The quantity of steam at 60 pounds pressure that will pass into the atmosphere in an hour through an orifice one square inch in area is estimated to be about 4,000 pounds. If the steam be wet the quantity wasted in the jet would be still greater. This size of steam jet is fitted in many ships whose engines indicate at full power about 1,000 I.H.P. Taking the consumption of steam per indicated horse-power in these engines, if compound, at 20 pounds per hour, the steam expended in the jet to produce the draught is equivalent to 200 I.H.P., or about one-fifth of the horse-power developed by the engines. The weight of steam in pounds per minute that would pass through an orifice one square inch in area, similar in form to the nozzle of a blast pipe, may be taken approxi-

mately to be about $\frac{13}{15}$ths of the absolute pressure in pounds per square inch. If the entrance to the orifice be square, as in the case of safety-valves, the weight in pounds per minute may be taken as three-fourths the absolute pressure. The steam blast can only be considered as a rough and ready method of increasing the power of the boilers, and attention has naturally been turned to the attainment of this end by mechanical means, which promise to be more economical and efficient.

The principal plans yet tried are:—

1. Fitting a centrifugal fan or exhauster in the funnel to draw off the products of combustion.

2. Blowing jets of compressed air into closed ashpits.

3. Admitting jets of compressed air into the base of the funnel in a similar manner to the steam jet.

4. Closing the stokehold and keeping it filled with compressed air.

The fitting of an exhauster in the funnel, through which the whole of the gases from the boiler must pass, is only practicable for small powers; but in those cases in which it has been applied very economical results have followed. In this system the power can be easily regulated, and the temperature of the gases leaving the funnel need not be greater than that of the steam in the boiler, so that the whole of the heat may be utilised if the heating surface be large enough and properly arranged. In most ships, however, the apparatus would require to be so large and unwieldy to admit of the whole of the gases passing through it at a sufficiently high velocity, that its application would be impracticable and inadmissible. This system, therefore, cannot be considered as a solution of the question of forced draught for marine boilers, and its advantages need not be further discussed.

The second plan, viz. blowing air into the ashpits, is an economical and efficient method of increasing the power

of boilers, and one that can be readily controlled; but there are difficulties attending its practical application on board ship for general use. It necessitates closed ashpits, as the pressure in the furnaces is greater than that in the stokehold, and unless proper precautions be taken before opening the furnace doors at the times of firing, the flame will be blown into the stokehold, with possibly dangerous consequences. It has, however, been used in the American Navy with considerable success for the burning of anthracite coal, for which a strong draught is necessary. If a good and reliable arrangement of mechanical firing were devised for marine boilers, which would render the frequent opening of the furnace doors unnecessary, probably this system would be the most efficient, as the supply of coal and air could be regulated to suit each other, and the most economical results would ensue. The air might also be heated before being blown into the ashpits, which would still further increase the economy of combustion. For the present, however, this plan must be regarded as unsuitable for general application to marine boilers.

The blowing of jets of compressed air into the base of the funnel has been tried to some extent in the French Navy, and the first reports of the experiments were favourable, but the actual results on board ship do not appear to have been altogether satisfactory, as the more recent French ships have not been fitted on this system.

The fourth plan, by which the stokeholds are made into airtight chambers and kept filled with compressed air by means of blowing fans, is now being generally applied to warships. This system was first adopted by the designers of the fast torpedo-boats, and very high powers indeed were obtained from the boilers of these boats when worked under air-pressure. In each of these boats only one boiler was fitted, so that the application of the system was more

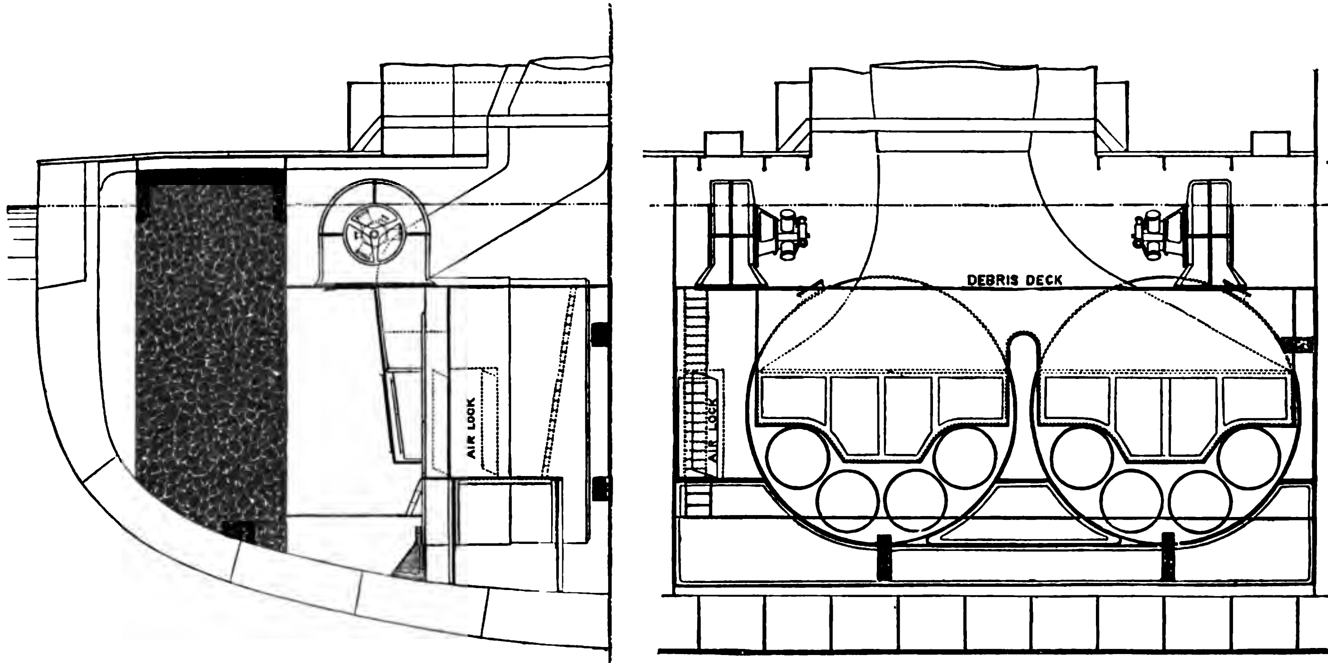

FIG. 14.

simple than in the case of vessels containing a number of boilers.

The general method in which this system is carried out in ships of the Royal Navy is shown in Fig. 14. The stokeholds are enclosed for the purpose of being placed under air-pressure by fitting a horizontal ceiling, about 10 or 12 feet above the floor plates, extending from the coal-bunker bulkheads to the fronts of the boilers, and from this ceiling vertical screen plates are carried down between and at the ends of the boilers to meet the front boiler bearers. These screen plates are worked around the fronts of the boilers to enclose the smoke-boxes so as to keep the stokeholds cool, and are carried back sufficiently far at the sides of the boilers to clear the water gauges. It will thus be seen that the stokeholds proper are readily made into closed airtight chambers of comparatively small dimensions. The screens are shown in thick lines on the diagrams.

Debris deck.—The ceilings of the stokeholds usually form portions of the debris or splinter deck, which is now fitted over all the openings in the machinery department, to protect the steam pipes and the people in the engine and boiler rooms from injury from fragments of shot or shell or other debris. On this debris deck the blowing fans for producing the air-pressure in the stokeholds are carried, one at each end of each stokehold if possible, so as to distribute the supply of air to the furnaces.

Air-locks.— In order to provide for passage to and from the stokeholds when under pressure air-locks are fitted. These consist of small airtight chambers fitted with two hinged doors, as shown in Fig. 15. In passing through, one door only is open at a time, which makes it possible to enter or leave the stokehold without causing the air to escape and its pressure in the stokehold to fall. A section through the air-lock is shown at A and a front view at B. Air-locks are necessary at all places at which

communication is made between the compartments under pressure and any other part of the ship.

To facilitate exit from the stokeholds in case of emergency, hinged doors which can be opened from the under side are fitted in the debris deck, and vertical ladders or rungs fitted to the bulkhead are carried from these openings to the stokehold plates.

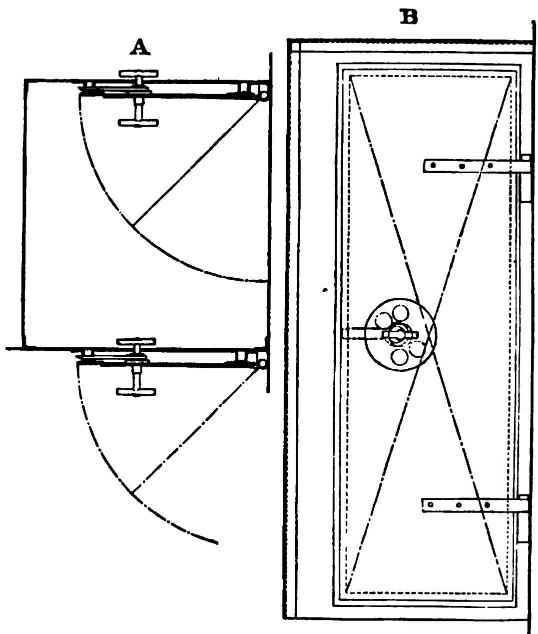

Fig. 15.

In the stokeholds of many of the fast cruisers no special horizontal ceilings are required to be fitted, as the deck of the ship covers the whole of the stokeholds. All that is necessary is to carry vertical screen plates around the boilers from the deck to the boiler bearers, so as to isolate the stokeholds. The other fittings are similar to those in

the armour-clad ships, modified in detail as required to suit the different arrangements of the ships.

This system of closed stokeholds and forced draught possesses many advantages for war ships. The most important feature is the great reduction that can be made in the space and weight required for the boilers; the extra power necessary for full-speed working, instead of being obtained by the provision of additional boilers which would occupy much space and weight, although but seldom required, is provided by the addition of fans and screens, which are comparatively inexpensive and involve very little additional space and weight.

The ventilating pipes with large reversible cowl-heads above the upper deck, necessary for natural ventilation, but which are generally objectionable and frequently interfere with the working of the ship or guns, can be to a great extent dispensed with, and the openings in the deck may be reduced to the minimum required for the supply of air to the fans and for convenient access to the engine and boiler rooms. The supply of air to the boiler-rooms being entirely provided by the fans, the power of the ship is practically independent of the force and direction of the wind, which is a matter of no small importance, especially in the tropics, and the power developed can be easily regulated by the speed at which the fans are driven. A most objectionable feature with the steam jet was the great noise it made in the funnel; but with the system of closed stokeholds and blowing fans no noise is made when working at full power, so that the deck operations are not interfered with.

The first ships in the Royal Navy to which this system was applied were the composite sloop 'Satellite' and the armour-clad turret-ship 'Conqueror.' The particulars of their steam trials were communicated to the Institution of Naval Architects by Mr. R. J. Butler, of the Admiralty, in

a paper read during the session of 1883. During the four hours' full-power trial of the 'Satellite,' with natural draught, 10·15 I.H.P. was developed per square foot of fire-grate. With an air-pressure in the stokeholds equal to $1\frac{1}{4}$ to 2 inches of water, 16·9 I.H.P. per square foot of fire-grate was obtained, being an increase of 66·5 per cent.

The fire-grates in the boilers of the 'Conqueror' were very long, and the results per square foot of grate during the trials, both with natural and forced draught, were therefore comparatively low. With natural draught 7·96 I.H.P. per square foot of fire-grate was developed, whilst with a mean air-pressure equal to $1\frac{3}{4}$ inches of water, the power was increased to 13·41 I.H.P. per square foot of grate, showing a gain of 68·6 per cent. A trial in basin, with shortened fire-bars, was subsequently made, during which the I.H.P. per square foot of fire-grate was 16·46, with an air-pressure equal to 2 inches of water.

During the three-hours' forced draught trial of H.M.S. 'Rodney,' made on June 13, 1885, the average I.H.P developed per square foot of fire-grate was 16·8, the air-pressure varying from $1\frac{1}{2}$ to 2 inches of water. During the last hour of the trial the air-pressure was kept equal to 2 inches of water, and the I.H.P. developed during this hour was equal to 17·3 I.H.P. per square foot of fire-grate. With natural draught, under the most favourable circumstances, these boilers could not be expected to develop more than 10 I.H.P. per square foot of fire-grate, so that the increase of power due to the forced draught may be estimated to be at least 70 per cent.

CHAPTER VI.

ARRANGEMENT AND EFFICIENCY OF BOILERS.

In this chapter we will consider more particularly the form and arrangement of the different types of marine boilers in general use, and their efficiencies as shown by actual performances.

In the first place, a little space must be devoted to the discussion of the properties and arrangement of the heating surface, which is perhaps the most important element to be considered in the design of a boiler, for it is obvious that the evaporative power of a boiler must depend mainly on the efficiency of its heating surface. The duty of the heating surface is to transmit the heat from the products of combustion to the water in the boiler; so that if the combustion be perfect the efficiency of the heating surface would be the efficiency of the boiler. The two terms must not, however, be taken as synonymous, for the efficiency of the boiler depends on the completeness of the combustion as well as on the efficiency with which the heat is transmitted.

The two efficiencies may be defined as follows:—

1. *Efficiency of the boiler* is the proportion borne by the heat transmitted to the total quantity of heat that would be yielded by the complete combustion of the fuel.

2. *Efficiency of the heating surface* is the proportion

borne by the quantity of heat transmitted to the water in the boiler, to that available for transmission.

The conditions on which the efficiency of the heating surface depend are :—

1. Its extent, nature, and condition as to cleanliness (scale, soot, &c.).

2. Its form, position, and arrangement.

3. The difference of temperature between the fluids in contact with the two faces.

4. The time allowed for the transmission of heat.

5. The nature of the medium for transmitting heat and the manner in which the heat is transmitted, whether from flame, incandescent fuel, or heated gas.

In comparing the evaporative powers of boilers it is not sufficient to estimate simply the total heating surface, consisting, as it does generally, of furnaces, combustion-chambers, tubes, &c., for the powers of transmission of these surfaces differ greatly.

In an experiment made by Mr. Armstrong, by placing a hot substance in the interior of a cubical metallic box submerged in water, it was found that the upper face generated steam more than twice as fast as the vertical sides, per unit of area, whilst the lower face yielded none at all. This was due to the difficulty with which steam separates from vertical surfaces to give place to fresh particles of water, so that a thin film of steam is formed in contact with the plates and acts as a non-conductor. It is evident that the steam could not escape at all from the bottom face. By slightly inclining the box, the elevated side gave off steam much more freely, so that its rate of evaporation was increased, whilst from the depressed side the steam escaped so slowly as to lead to an over-heating of the metal.

In an ordinary boiler the crowns of the furnaces form the most efficient heating surface, next come the sides of

the furnaces above the fire-bars, then the tops of the combustion-chambers, then the sides and ends, and last of all the tubes.

By the use in boilers of long small tubes, through which the heated gases have to pass, a large amount of heating surface can be obtained in a small space, and this arrangement is necessary in the present type of marine boilers, though heating surface in this form is comparatively inefficient. Only the upper halves of the tubes can be considered as effective heating surface, owing to the difficulty with which the steam can detach itself from the lower halves, and also in consequence of the soot, &c., deposited inside the tubes. The direction of the tubes is necessarily the same as that of the currents of hot gases on their way to the funnel, instead of being normal to it, as it should be in order to extract the maximum amount of heat from the gases. Flame cannot pass through long tubes of small diameter, and consequently the useful combustion of the gases cannot extend much beyond the combustion-chamber. The flame is extinguished within a few inches from the entrances of the tubes, and the gases pass through unconsumed, possibly, in cases where strong blast is used, to burst into flame in the uptake or funnel.

Supposing the gas entering the tubes to be of uniform temperature, the manner in which the heat is transmitted to the water is somewhat as follows:—The particles impinging on the upper surfaces of the tubes give up some of their heat, and thus becoming heavier descend, the motion of the current carrying them forward in their descent, so that their places are taken by the hot particles below and behind them, which act in a similar manner. In all probability, owing to the great velocity of the gases through the tubes, a large proportion of the hot gases in the centre and lower parts of the tubes never

reach the surface at all, so that their heating effect is entirely lost.

It is evident, from the nature of the action of the gases in the tubes, that the efficiency of vertical tubes would be very small. Boilers with vertical tubes are not used for marine purposes, except for winches or other auxiliary machinery, for which they are sometimes convenient, but they are often fitted in iron works to utilise the waste heat passing off from the blast furnaces. The gases from the furnaces are conveyed to a flue passing up the centre of a very high cylindrical boiler, so as to transmit heat to the water surrounding the flue. In order, however, to render these boilers efficient, it is necessary to fit transverse water tubes across the main flue to break up the current of gas and extract the heat from it. A front view and section of a boiler with vertical central flue and cross tubes is shown in Fig. 16.

In horizontal tubes the first few inches of length are the most efficient. In coming in contact with the first unit of length, the gases part with some of their heat and proceed at a reduced temperature, which is further reduced as the gases pass along the tubes. The intensity of the draught would also draw the flame a few inches into the tubes, and thus considerably increase their evaporative power at the entrances. It is clear from these considerations that only a comparatively small evaporative power can be expected from the exit ends of long tubes, aud this is confirmed by experiment. In 1830, Stephenson found that in a locomotive boiler open to the atmosphere, with the fire-box separated from the barrel, one square foot of fire-box was equal to three square feet of tube surface. In 1840, experiments made by dividing the barrel of a locomotive boiler into six compartments, that next the fire-box being six inches and the others twelve inches long, showed that the first six inches of tube surface were equal,

area for area, to the fire-box surface, the second compartment was only one-third as effective, whilst in the remaining compartments the rate of evaporation was small.

In 1864 further trials were made on a multitubular boiler five feet long, the tubes being divided into six parts by plates at intervals. The compartment next the fire-box was only one inch long, the second ten inches, and

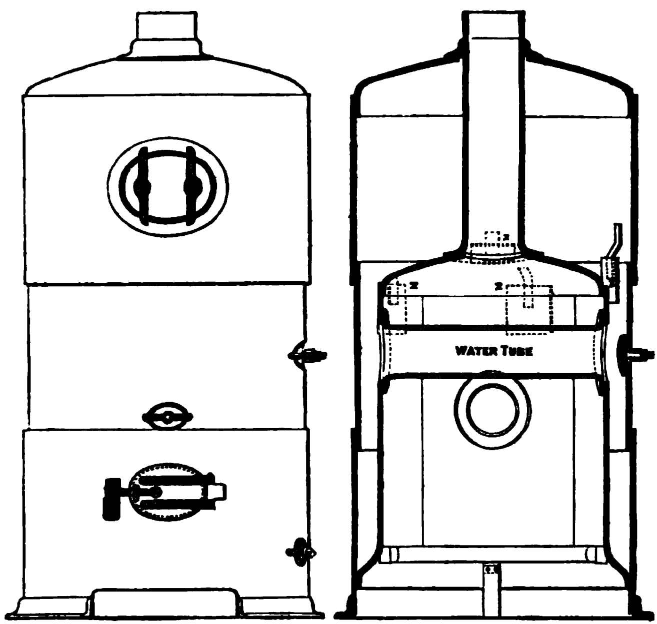

FIG. 16.

the four remaining were each twelve inches in length. The following quantities of water were found to have been evaporated after three hours' work :—

Compartment	No. 1	(1 inch long)	. . . 46 ozs.
„	2	(10 „)	. . . 47 „
„	3	(12 „)	. . . 30 „
„	4	(„ „)	. . . 22 „
„	5	(„ „)	. . . 18 „
„	6	(„ „)	. . . 17 „

The very high rate of evaporation in the first compartment, which was only one inch long, was no doubt, to a great extent, due to the action of the tube plate, but a comparison of Number 2 compartment with the others shows how rapidly the evaporative efficiency of the tubes diminished as the gases passed from the combustion-chamber to the smoke-box.

In ordinary practice the diminution would probably not be so great as that shown in this experiment, because the strong draught in an actual boiler, when at work, would draw the flame for some distance into the tube, and thus increase its evaporative power.

For coals that burn with a long flame, the diameter of the tubes should be large, so as to allow the flame to pass as far along the tubes as possible. But for coke and anthracite coal, if the hydrocarbons and carbonic oxide can be burnt before they reach the tubes, they may be made of small diameter, so as to increase the surface and facilitate the action of the hot gases on it.

As a general rule the ratio of length to diameter of tube in marine boilers rarely exceeds 30 to 1, and the area through the tubes should be about one-seventh the grate area. In locomotive boilers the ratio of length to diameter of tubes is often as great as 120 to 1.

For a given description of boiler the evaporative efficiency will depend mainly on the ratio between the quantity of coal burned and the extent of heating surface to transmit the heat of combustion to the water.

The relations between grate area, heating surface, and consumption of coal and water in locomotive boilers have been carefully investigated by Mr. D. K. Clark, who arrived at the following conclusions, which are probably approximately correct for marine tubular boilers.

1. For a given extent of heating surface, the economical hourly consumption of fuel or water decreases

directly as the grate area is increased, and consequently, in order to maintain the same efficiency or economical effect, the total hourly consumption should be decreased at the same rate as the grate area is increased.

2. For a given grate area the total hourly consumption should vary as the square of the heating surface; that is, if the area of heating surface be doubled, four times the quantity of fuel can be burned with the same grate area, and the same evaporative efficiency or economy maintained.

3. For a given hourly consumption the area of fire-grate should vary as the square of the heating surface in maintaining the same efficiency. That is, if the heating surface be doubled, the grate area may be increased four times, and the same economical consumption maintained.

The first of these conclusions probably holds good for all boilers. In general it may be said that there cannot be too little grate area for economical evaporation. The speed of evaporation in boilers is, however, of great importance, and this will not generally allow the reduction of the grate surface to the point that would give the greatest evaporative economy. In all boilers there is a certain rate of combustion per foot of grate which is the maximum consistent with economy, and the grate area must be made large enough for the rate of evaporation required, the amount of steam produced per hour being roughly proportional to the quantity of coal burned in the same period.

From the second and third conclusions it would appear that when the heating surface is doubled the economical hourly consumption may be increased fourfold either by increasing the rate of combustion or the size of the fire-grate. The evaporative power of the boiler, therefore, increases in a greater ratio than the heating surface whilst the same efficiency is maintained.

This is, however, modified by the consideration that there is a maximum quantity of coal that can be economically burnt on each square foot of fire-grate, which limits the power to be derived from increasing the heating surface whilst the fire-grate remains constant, and, as before pointed out, the area of the fire-grate is restricted by practical conditions.

Rankine has given the following approximate formula for calculating the efficiency of a boiler :—

Let E denote the theoretical evaporative power of the coal.
E′ „ its available evaporative power.
S „ the number of square feet of heating surface per square foot of fire-grate.
and F „ the number of pounds of coal burnt per square foot of fire-grate per hour.

Then the efficiency of the boiler is

$$\frac{E'}{E} = \frac{B\,S}{S + A\,F}$$

Where B and A are constants to be determined by experiment.

The fraction on the right-hand side of the equation, if B be omitted, represents the efficiency of the heating surface itself. B is a fractional multiplier to allow for miscellaneous losses of heat whose value is found by experiment. A is a constant to be found empirically, and is probably proportional approximately to the square of the quantity of air supplied per pound of coal.

Rankine gives the following values for the constants A and B deduced from the practical performances of a number of boilers.

	B	A
I. The convection taking place in the best manner, either by introducing the feed water at the coldest part of the boiler and making it travel gradually to the hottest; or by heating the feed water in a set of tubes placed in the uptake. Draught produced by the chimney only	1·0	0·5
II. Ordinary convection and chimney draught only .	$\frac{11}{12}$	0·5
III. Best convection and forced draught	1·0	0·3
IV. Ordinary convection and forced draught . .	$\frac{19}{20}$	0·3

When there is a superheater or feed-water heater its surface must be included in computing S.

This formula is framed on the assumption that the losses from imperfect combustion and excess of air are inappreciable, and that the construction and management of the furnaces are the best possible. If this be not the case, the coefficients A and B must be modified to suit the altered circumstances.

Case II., viz. that of ordinary convection and chimney draught, is that of the majority of marine boilers. Rankine gives the value of B in this case as $\frac{11}{12}$, and this appears to agree very well with the actual results of the performances of the high rectangular boilers working with steam pressure of about 30 pounds per square inch. Mr. Robert Wilson gives $\frac{4}{5}$ as the value of B, which seems to approximate more closely to the performances of cylindrical marine boilers.

Feed-water heater.—The feed-water heater, in addition to increasing the economy of working, also appears to be conducive to the preservation of the boiler to which it is attached. This is probably, to some extent, due to the uniformity of temperature at which the boiler can be maintained by supplying it with hot feed-water, but mainly to the action of the water-heater in expelling the air from the feed water, the material of the feed-water heater itself being attacked instead of the boiler plates,

stays, etc. Feed-water heaters are more frequently fitted to land than to marine boilers.

Low-pressure boilers.—Figs. 17 and 18 represent the most general type of marine boiler for steam pressures not exceeding from 30 to 40 pounds per square inch above the atmosphere. On account of the higher steam-pressures now generally adopted for new ships, it is becoming obsolete, its form not being suitable for high pressures, as in these boilers the strains are resisted principally by the action of straight iron stay-rods. For pressures above 30 or 40 pounds per square inch, these would have to be so numerous and closely spaced, that the boilers would be excessively heavy, and the internal parts inaccessible.

The furnaces are made with flat sides, and from their shape it is easily seen that, whatever their width may be, arrangements can generally be made to keep the crowns of the furnaces sufficiently high above the bars to allow the gases to mingle freely with air, whilst the bottoms of the ashpits may be kept low enough to permit an ample supply of air to pass through the fires for combustion. No doubt the good results that have been obtained from these boilers have been due to a great extent to the very roomy furnaces and combustion-chambers with which they are generally fitted.

In these, as in most other boilers, the furnace doors are made considerably smaller than the furnaces themselves, so that when they are opened for firing, &c., the quantity of cold air admitted may be as small as possible.

The furnaces end in a spacious combustion-chamber, in which the combustion of the gases should be as completely effected as possible. The furnaces are usually arranged in pairs, each pair having a common combustion-chamber, as shown in the diagrams. The object of this was pointed out at page 50. It is not advisable in general

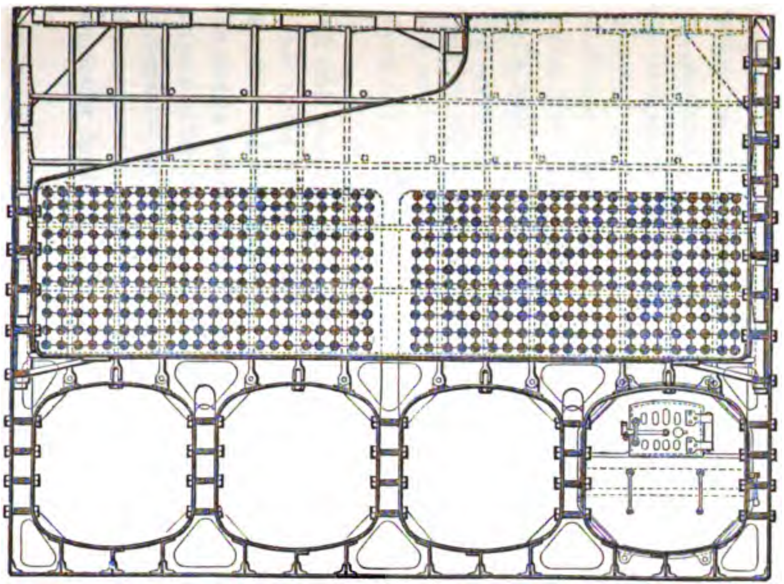

Fig. 17.

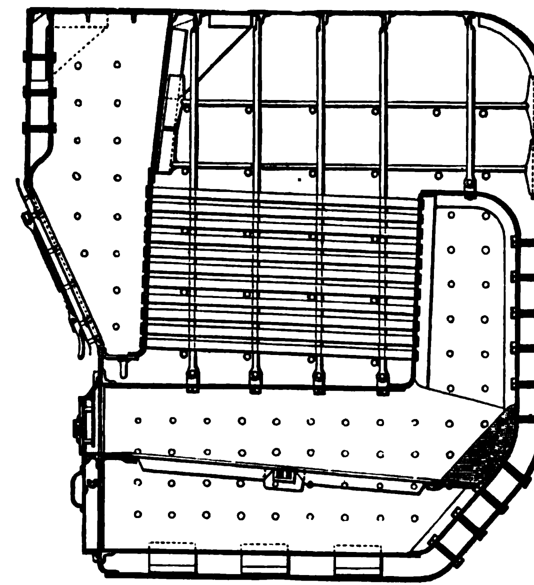

Fig. 18.

to allow more than two furnaces to enter the same chamber, because the gases coming from the different furnaces may cause a confusion of currents and check the draught. In addition to this it has been found that when several furnaces lead into the same chamber the draught through some of the fires is often greater than through others, so that the work done is unequally distributed and the performance of the boiler less efficient.

We may add in passing that the same principle applies to the case of several boilers, the uptakes from which lead into the same funnel. In this case diaphragms should be carried to a sufficient height up the funnel to keep the currents from the different boilers separate until they are all moving in the same direction, when no *shock* will result from their mingling.

From the combustion-chamber the smoke and gases pass through tubes, arranged over the furnaces, to an uptake in the front of the boiler, in which they all unite, and are conveyed to the funnel.

Many boilers of this type are fitted with superheaters, which are described in Chapter VII. The heat added to the steam in the superheater from the hot gases on their way to the funnel would otherwise have been wasted, and there can be little doubt that much of the saving that has resulted from superheating steam has been due to the partial utilisation of the waste heat in the escaping gases.

The plates in these boilers are usually very thin for the pressure of steam, the necessary strength being obtained by suitable staying. The area of the stays is calculated on the assumption that they have to resist the action of the whole of the steam-pressure. In arranging the stays care should be taken to place them sufficiently close together to prevent any appreciable buckling of the plates between the stays when the pressure is acting on them, which would

tend to make the plate open off the stay at the stay-hole and cause local weakness.

If we take an average of the performances of the best examples of boilers of this type we find that, at full power, about 30 pounds of coal are burnt per hour, and 10 indicated horse-power developed, per square foot of fire-grate. If these boilers are fitted with superheaters and the engines with surface condensers, as has generally been the case, the consumption of feed water per hour may be taken at about 26 lbs. per indicated horse-power.

Therefore the quantity of water evaporated per pound of coal is equal to

$$\frac{10 \times 26}{30} = 8 \cdot 7 \text{ lbs.}$$

Taking the theoretical evaporative power of the coal as 14·5 lbs. from 100° Fahr. at 275° Fahr., which is the temperature corresponding to a pressure of 30 pounds per square inch above the atmosphere, this gives as the actual efficiency of the boiler,

$$\frac{8 \cdot 7}{14 \cdot 5} = 0 \cdot 6.$$

These boilers have about 30 square feet of total heating surface per square foot of fire-grate, so that their efficiency calculated by Rankine's formula would be

$$\frac{E'}{E} = \frac{B\,S}{S + A\,F}$$
$$= \frac{\frac{11}{12} \times 30}{30 + \frac{30}{2}} = \frac{11}{18} = 0 \cdot 61.$$

In this case, therefore, the results of the actual performance agree with those calculated by Rankine's formula.

Figs. 19, 20, 21, and 22 show the type of boiler that was generally used before the adoption of high steam-pressures, for gun-vessels and sloops of shallow draught, in which the depth of hold was not sufficient to admit of

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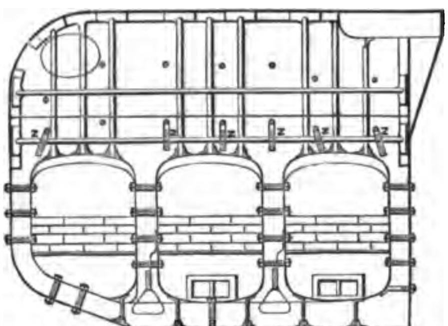


FIG. 19.

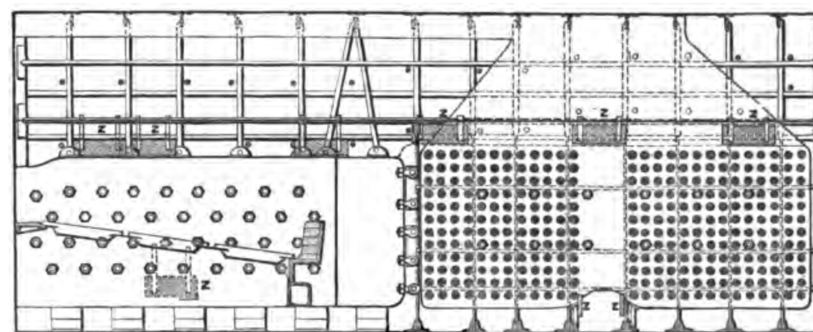


FIG. 20.

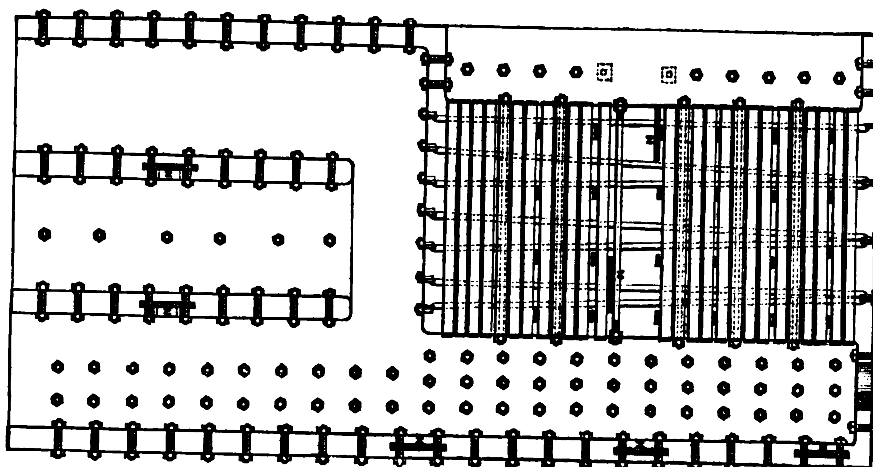


Fig. 21.

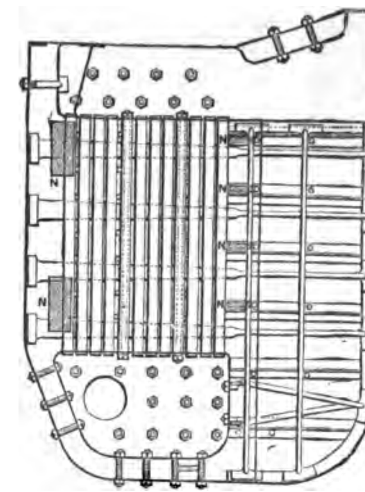


Fig. 22.

the tubes being placed over the furnaces, it being necessary to keep the whole of the machinery and boilers below the water-line. It is not a particularly good type, but it was a necessary one. It will be seen from the diagrams that the top row of tubes is just level with the crowns of the furnaces. The zinc slabs fitted for preservation of the boilers are marked Z.

There are various arrangements of this type of boiler. In the diagrams there are three furnaces at one end of the boiler, which unite in a common combustion-chamber running along one side of the boiler, and the tubes pass from the chamber, at right angles to the direction of the furnaces, to the smoke-box at the opposite side. The many different ways in which this type may be modified to suit various circumstances can be easily seen.

A number of twin screw gun-vessels of the 'Plover' and 'Seagull' class were fitted with boilers of this description pressed to 25 pounds per square inch. The engines were fitted with ordinary jet injection condensers, and for these engines a consumption of feed water of about 34 pounds per indicated horse-power per hour may be taken as a fair average. On the full power trials these boilers developed from 7 to 8 indicated horse-power per square foot of fire-grate. Taking the consumption of coal per indicated horse-power per hour at 4 pounds, which is probably below rather than above the average, this would give the consumption of coal, in the best examples, as about 32 pounds per square foot of fire-grate per hour.

The water evaporated per pound of coal would thus be equal to

$$\frac{34 \times 8}{32} = 8{\cdot}5 \text{ lbs.}$$

and the actual efficiency of the boiler would therefore be

$$= \frac{8{\cdot}5}{14{\cdot}5} = 0{\cdot}58.$$

The total heating surface in these boilers is about twenty-eight times the grate surface, so that the efficiency by Rankine's rule is

$$\frac{E'}{E}=\frac{\frac{11}{12}\times 28}{28+\frac{32}{2}}=\frac{7}{12}=0\cdot583.$$

In these boilers, unless proper arrangements be made, the lower rows of tubes will be much less efficient than the upper rows, so that the efficiency of the boiler will be decreased. The gases when they leave the furnaces make direct for the funnel through the upper rows of tubes, very little comparatively passing through the lower rows, which are in addition more liable to become choked by the deposit of soot, &c. In some cases this has been remedied to a certain extent by fitting brick deflecting bridges to compel the gases to pass to the bottoms of the combustion-chambers before entering the tubes.

For the higher pressures of steam now commonly used, the rectangular forms of boilers just described, or, as they are often called, 'box boilers,' are not suitable, and boilers with cylindrical shells and furnaces have been substituted for them.

High-pressure boilers.—These may be divided into two classes, high boilers and low boilers, in the same way as the flat-sided boilers for the lower pressures; the low boilers being fitted in vessels of small depth of hold to keep them below the water-line for protection from shot, &c.

Figs. 23 and 24 show the general arrangement of the high type of cylindrical marine boiler used for pressures of 60 pounds per square inch and upwards. Its general features are similar to those of the type of boiler shown in Figs. 17 and 18. In its details, however, it is very different, and it is inferior to the low-pressure boiler both in evaporative power and in economy of generation of steam. In the diagram each furnace is shown as having a separate

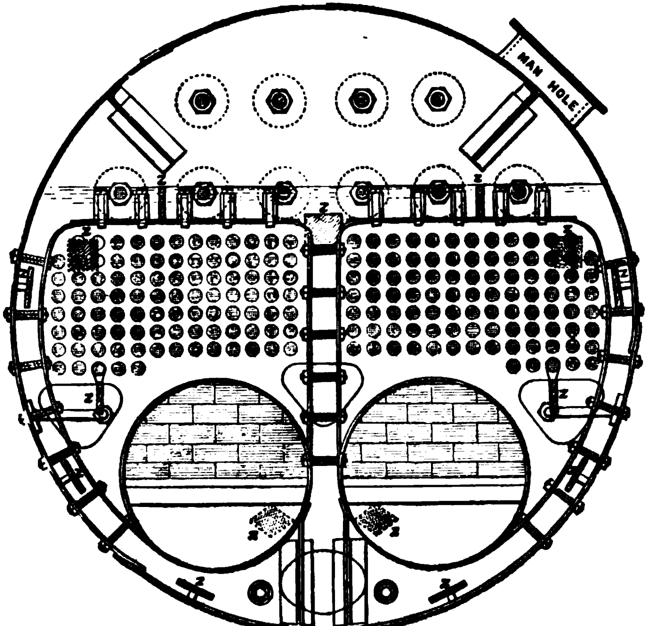


Fig. 23.

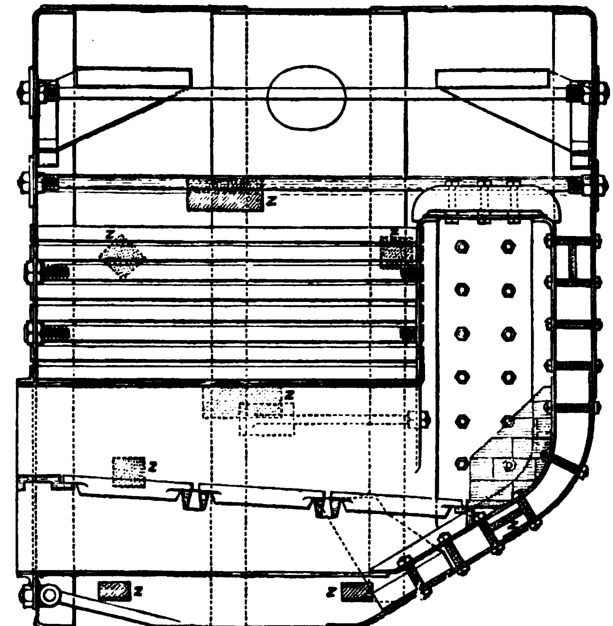


Fig. 24.

combustion-chamber, which is the most general arrangement in these boilers, though sometimes the two furnaces enter a common chamber.

The comparatively defective evaporative power and economy in these boilers is mainly due to the form of the furnace. In the low-pressure boilers, with furnaces of approximately rectangular forms, the necessary distance above and below the fire-bars can generally be obtained whatever the width of the furnace may be. But in the high-pressure boilers, in which the furnaces are cylindrical, it is obvious that the height above and depth below the bars are no longer arbitrary, but are dependent on the diameter of the furnace. With this form of furnace it is difficult, in most cases, to keep the crown of the furnace sufficiently high above the fires, or to make the ash-pit large enough to allow an ample supply of air to the fires. The smaller the diameter, and the greater the length of the furnaces, the greater will be their inefficiency. The length of the cylindrical furnace is rarely allowed to exceed seven feet, and the diameter is usually made as large as possible, to reduce the loss from imperfect combustion.

In these boilers the front itself forms the outer tube plate, and the smoke-box and uptake form an entirely external fitting instead of being partly built in the boiler, as in the case of low-pressure boilers.

Superheaters are rarely fitted to high-pressure boilers, because the greatest temperature of superheated steam that can be safely used in ordinary marine engines, so far as present experience goes, appears to be about 360° to 370° Fahr., so that there is very little margin for superheating steam of 60 pounds pressure and upwards.

In some of the larger ships the boilers, instead of being circular in section, with two furnaces, as shown in Figs. 23 and 24, are fitted with three furnaces, and the shells have

semicircular tops and bottoms, with straight sides between, strengthened with angle irons and stay-bars, as shown in Fig. 25. In several recent ships large cylindrical boilers with four furnaces in each have been fitted as shown in Fig. 14.

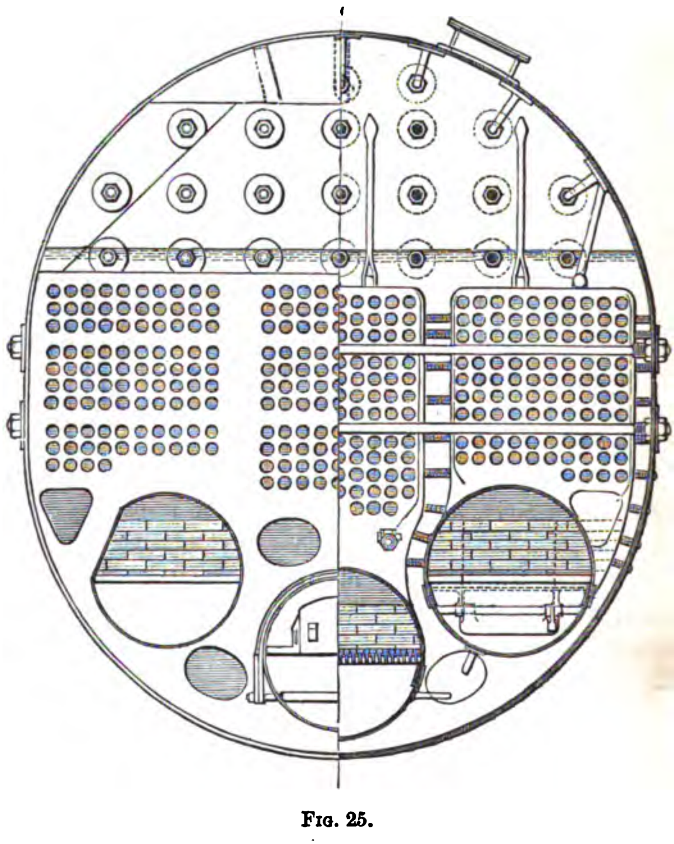


Fig. 25.

The average performance of boilers of this type with natural draught may taken to be as follows:—

Coal burnt per hour per sq. foot of grate	21 lbs.
Indicated horse-power developed per sq. foot of grate .	8·5.

The coal burnt per square foot of grate per hour is therefore only about seven-tenths of the quantity burnt in the low-pressure boilers shown in Figs. 17 and 18. The engines worked by these boilers are generally of the compound type and the consumption of feed water per hour may be taken at about 20 pounds per indicated horse-power. The quantity of water evaporated per pound of coal is therefore

$$=\frac{8\cdot5 \times 20}{21}=8\cdot1 \text{ lbs.}$$

instead of 8·7 lb. as in the high box boiler. Taking 14·4 as the evaporative power of the coal for a pressure of 60 pounds per square inch, this gives for the actual efficiency of the boiler,

$$\frac{8\cdot1}{14\cdot4}=0\cdot56.$$

This shows that cylindrical boilers of this type are, in themselves, neither economical nor rapid generators of steam. The economy that has resulted from their use has been entirely due to the increased expansive working of the steam, as there is a loss in the boilers themselves.

The ratio of the total heating surface to grate surface in these boilers has been generally about 25 to 1, so that the efficiency computed by Rankine's rule would be

$$\frac{E'}{E}=\frac{\frac{11}{12} \times 25}{25+\frac{21}{2}}=0\cdot646.$$

It appears, therefore, that the value of B given by Rankine should be modified for cylindrical boilers. By taking the value of B to be $\frac{4}{5}$, as given by Mr. Robert Wilson, we find the theoretical efficiency of these boilers to be,

$$\frac{E'}{E}=\frac{\frac{4}{5} \times 25}{25+\frac{21}{2}}=0\cdot563.$$

which agrees much more nearly with the actual efficiency. In the more recent boilers designed for forced draught

working the ratio of heating to grate surface has been considerably increased to make them suitable for the increased rate of combustion in the furnaces.

Another fact which tends to reduce the efficiency of cylindrical boilers is the often restricted area of water surface and volume of steam chest, which renders them more liable to priming. *Priming* is the technical name for the passage of water, with the steam, from the boilers to the engines, which sometimes takes place, and is liable to produce serious results, reducing the power of the engines, and causing severe strains on the cylinders, &c., and when it is excessive it may lead to overheating and burning of the boiler plates and tubes, by withdrawing a large quantity of water from the boilers.

Many cylindrical boilers have been fitted with large steam domes in order to increase the steam room. When this is not the case the depth of the steam chest should be at least from one-quarter to one-third the diameter of the boiler. In measuring the steam room the water level should be taken at about nine inches above the highest part of the heating surface.

The steam dome is an undesirable fitting in high-pressure boilers, and it is preferable to obtain sufficient volume of steam chest by increasing the diameter of the boiler if possible. If the size of the dome be large enough to materially increase the steam room, the cutting of the requisite hole in the shell of the boiler must reduce its strength at that part. If a steam dome be necessary in any case, it is advisable to construct it if possible with a contracted neck of stout material, having broad flanges to compensate for the material cut out. Steam domes are rarely fitted to marine boilers. Sometimes, however, they may be useful for affixing the boiler mountings to prevent a number of holes being cut in the shell of the boiler, and in these cases the edge of the hole cut in the shell plate

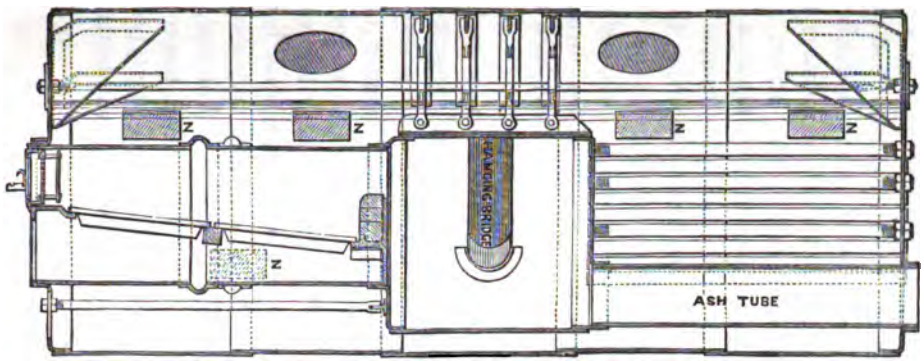


Fig. 26.

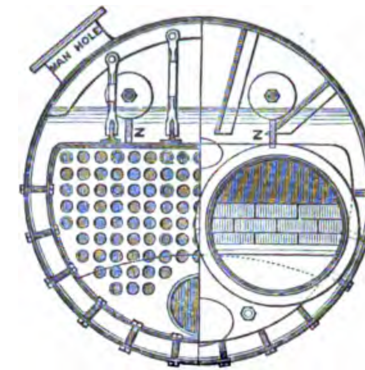


Fig. 27.

for the dome should be strengthened by having a wide stiffening ring riveted on.

The low type of cylindrical boiler shown in Figs. 26 and 27, which is fitted on board ships of the smaller classes, such as sloops, gun-vessels, &c., and also in several of the fast cruisers, have generally given much higher evaporative powers than those just described.

In the diagrams only two furnaces are shown, but in many recent ships boilers of this type with three furnaces have been fitted. The furnaces unite in a common combustion-chamber, and the tubes, instead of returning over the furnaces to an uptake in front of the boiler, are continued in the same direction as the furnaces to an uptake fitted at the opposite end of the boiler. In the combustion-chamber there is usually fitted a brick bridge, which causes the gases from the furnaces to go to the bottom of the chamber before entering the tubes, and renders the lower rows more efficient than they otherwise would be. It is probably owing to this, added to the more effective action of the crowns of the furnaces, that the results obtained from these boilers have been so satisfactory with respect to evaporative power.

In some of these boilers as much as 12 I.H.P. has been developed per square foot of fire-grate surface, with the steam blast. We may take as an average result of the best boilers of this type that 27 pounds of coal can be burnt per hour, and 11 indicated horse-power developed, per square foot of grate. Taking the consumption of feed water at 20 pounds per indicated horse-power per hour, we find that in these boilers only about 8·1 pounds of water are evaporated per pound of coal, so that while the speed of evaporation is increased, the economy is not.

The comparative results of the four types of boilers are given in the following table, which will be found convenient for reference:—

		Boiler No. 1	Boiler No. 2	Boiler No. 3	Boiler No. 4
Weight per I.H.P. including water	cwts.	1·50	1·85	1·60	1·65
I.H.P. developed per sq. ft. of grate	No.	10·0	8·0	8·5	11·0
Total heating surface per sq. ft. of grate	sq. ft.	30	28	21	27
Coal used per sq. ft. of grate per hour	lbs.	30	32	21	27
Water evaporated per lb. of coal	lbs.	8·7	8·5	8·1	8·1
Actual efficiency of the boiler		0·60	0·58	0·56	0·56
Efficiency calculated by Rankine's rule		0·61	0·583	0·646	0·61
Efficiency calculated by using Wilson's coefficient		0·53	0·51	0·563	0·561

Double-ended boilers.—Figs. 28 and 29 show a double-ended boiler. This type has been largely adopted for mercantile steamers and is also fitted in several ships of the Royal Navy. It is practically equivalent to two single-ended boilers placed back to back, but is not so heavy for equal power, because the weights of the end plates and of the water in the spaces at the backs of the combustion-chambers are saved.

In some of these boilers all the furnaces enter a common combustion-chamber. Generally, however, the combustion-chamber is divided by longitudinal vertical water spaces so that each opposite pair of furnaces has a common combustion-chamber, and this appears to be on the whole a very convenient and satisfactory arrangement. In boilers that have four furnaces at each end, as shown in the diagram, there are often two combustion-chambers only, two pairs of furnaces at opposite ends of the boilers discharging into one combustion-chamber. In many cases the combustion-chamber is divided into two parts by a transverse water space, so that the furnaces at opposite ends of the boiler are entirely divided from each other.

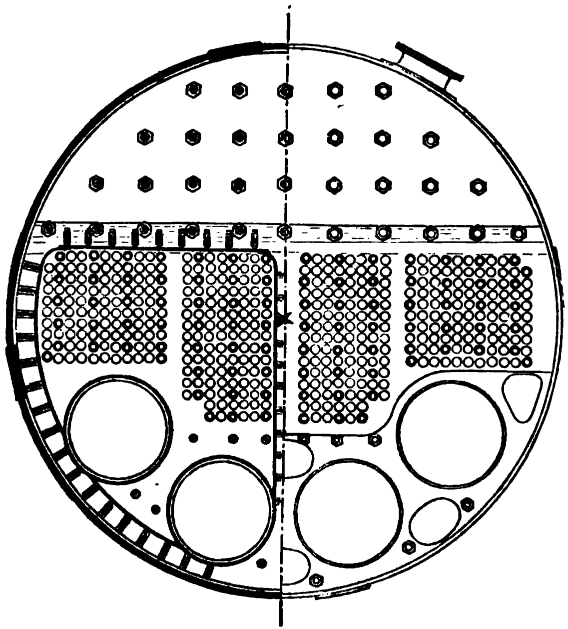


FIG. 28.

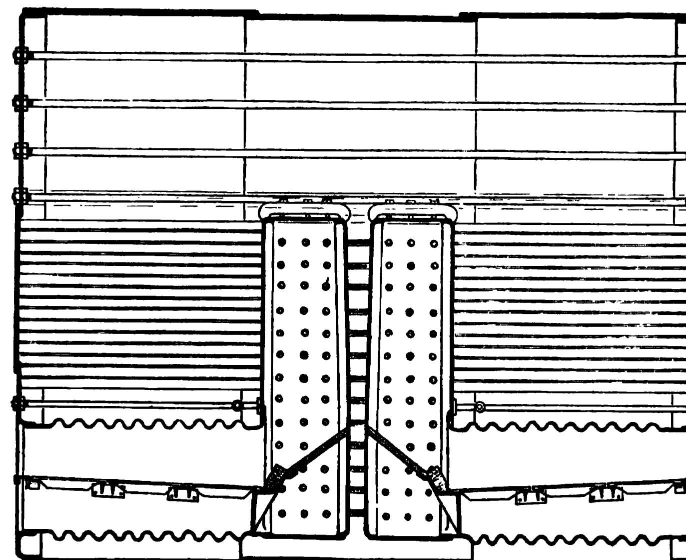


FIG. 29.

This plan has some advantages, especially for forced draught working, but it makes the boiler longer and heavier, and the water space in the centre with flat sides has to be closely stayed, which interferes with the cleaning of the boiler.

Locomotive boilers.—Figs. 30 and 31 illustrate the locomotive type of boiler which has been used in some special cases for marine purposes, such as the fast torpedo-boats, &c., in which the working pressures of steam have been from 120 to 150 pounds per square inch. In this type of boiler there is a broad deep fire-box at one end, the crown of which is strengthened by means of girder stays, as shown in the sketch. The furnace door is small, but the fire-bars are placed over the whole area of the fire-box, the air for the combustion of the coal being supplied from underneath, and it is clear from the form of the fire-box that there is considerable space and height above the fires to allow for the combustion of the gases. The barrel of the boiler is cylindrical, and contains the tubes which pass to a smoke-box at the opposite end of the boiler. In the cases in which these boilers have been employed the stokeholds have been closed in, and kept under a pressure of air, equal to about 3 or 4 inches of water, by means of blowing fans, to force the draught, the rate of combustion of coal per square foot of grate being from 60 to 70 pounds per hour.

Water-tube boilers.—We have already pointed out that tubes, when the heated gases pass through them, do not form a very effective heating surface, because the direction of the motion of the gases is parallel to the surface of the tubes instead of being normal to it. To remedy this, boilers have been constructed in which the water is contained in the tubes and the heated gases pass around them. The general arrangement of this type of boiler is shown in Figs. 32 and 33, and it will be seen that more space is given for the combustion of the gases, and the direction of

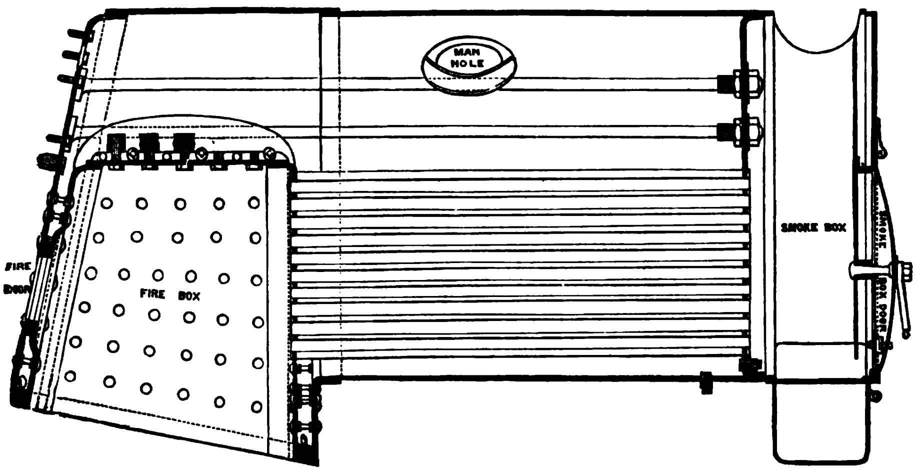


FIG. 30.

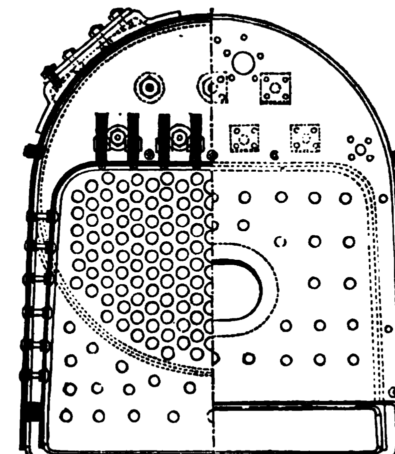


FIG. 31.

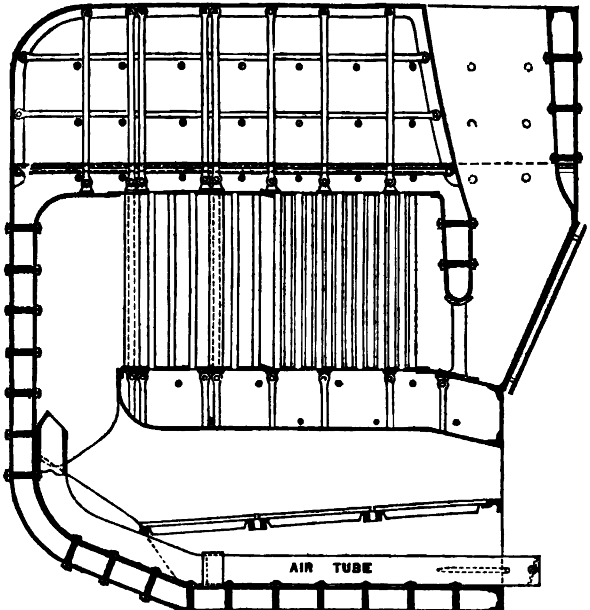


Fig. 32.

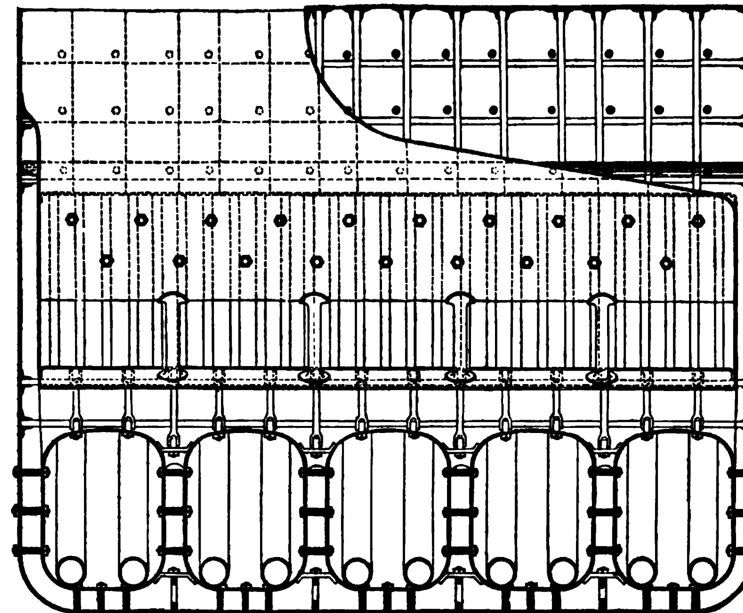


Fig. 33.

motion of the gases from the furnaces to the smoke-box is normal to the tubes, so that the heat is more completely absorbed by the water in the boiler.

These boilers gave economical evaporative results when the tubes were clean and the fires were not forced, but practical objections in the shape of difficulty of cleaning and of repair prevented their adoption. It was also found that when the fires were forced there was great tendency to priming, and in one ship this defect was so serious, that the boilers, although new, had to be taken out and ordinary tubular boilers substituted for them. The cleaning of these tubes from soot, &c., would take from four to five times as long as the sweeping of the tubes of an ordinary boiler, and if salt water were used for feed, or allowed to get in the boiler in any considerable quantity, which is almost necessarily the case in ordinary marine boilers, the tubes would probably become choked with scale and burnt, and any accident to one of these water-tubes whilst the boiler was at work would be much more difficult to remedy than in the case of an ordinary fire-tube boiler.

Independently of these defects, however, the time for these boilers may be considered as past, for, from the form of the shell, they are only suitable for what are now considered low pressures of steam; and it will only be in special cases that any boilers of the rectangular type, whether fire or water tube, will be constructed in future. For what are now the ordinary working pressures, the cylindrical boiler, either of the high or low type, appears to be the only form available, notwithstanding the objections already noted as to its comparative inefficiency of furnace, and its restricted area of water surface and volume of steam chest. In some special cases, the locomotive type described at p. 95 will probably be adopted. This is the case in the fast torpedo-boats, but it is doubtful if it will be employed for marine purposes generally.

Tubulous boilers.—For higher pressures of steam than those now used, it may perhaps be necessary to have boilers entirely built up of tubes, which are generally known as *tubulous boilers*. These have not yet been tried to any extent for marine purposes, but have been introduced in some special cases. Experience with them has so far not been altogether satisfactory, several serious explosions having occurred, in some cases unfortunately attended with loss of life. Possibly in many of these examples the failure might be traced to preventible causes, such as faulty construction or management, and not to defects inherent to the type of boiler.

It is easy to see that a boiler consisting entirely of small tubes, if proper care be exercised in the design of the details, is the strongest form that can be devised, and it is possible that as experience in their construction and management is gained, boilers worked with very high pressures may be introduced with safety and economy.

Mr. Loftus Perkins has made some boilers of this type for marine purposes that work at a pressure of 300 to 500 pounds per square inch with perfect safety, the general arrangement being shown in Figs. 34 and 35. They are made entirely of small iron tubes, butt-welded, of $2\frac{1}{4}$ inches internal, and 3 inches external diameter. The steam-collecting tube at the top of the boiler is of 4 inches internal, and $5\frac{1}{2}$ inches external diameter. The horizontal tubes are welded up at each end $\frac{1}{2}$-inch thick, and connected by small vertical tubes of $\frac{7}{8}$-inch internal and $1\frac{5}{16}$ inches external diameter. The fire-box is formed of tubes bent into a rectangular shape, placed $1\frac{3}{4}$ inches apart, and connected by numerous small vertical tubes $\frac{7}{8}$-inch internal diameter. The body of the boiler is made of a number of vertical sections, composed each of eleven tubes connected at each end by a vertical one. The vertical sections are connected at both ends by a vertical tube to the top ring

of the fire-box, and by another to the steam-collecting tube. The whole of the boiler is surrounded by a double casing of thin sheet iron, filled up with vegetable black to prevent loss of heat from radiation.

One great difficulty with tubulous boilers is to provide for the circulation of the water and the free escape of steam. The area through the vertical tubes in Perkins' boiler is stated to be sufficient for this, and also to prevent

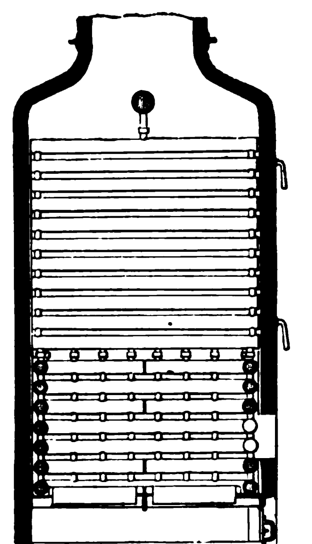


FIG. 34.

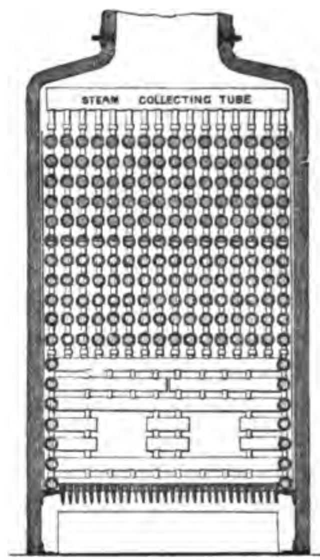


FIG. 35.

injury from the over-heating of the tubes in contact with the flame. Probably the introduction of the feed water into the lower rows of tubes in this boiler assists the circulation by rendering unnecessary the descent of water to supply the place of the steam generated.

It is necessary that these boilers should be supplied with absolutely pure water, as any deposit on the tubes would be not only injurious but dangerous. They are,

therefore, in the first instance, filled with distilled fresh water, and every precaution is adopted to prevent waste of water. All the boiler mountings must be kept perfectly tight, the safety-valves are screwed down to considerably more than the ordinary working pressure, and the surface condenser tubes are screwed and made absolutely tight at the ends to prevent the possibility of any sea-water getting in. The slight waste that does occur is made up by the addition of distilled fresh water, or, if this is not available, by doubly distilled sea-water, a special still, worked by a steam coil, being supplied for this purpose. In a seagoing ship an additional boiler fed with sea-water would be necessary for distilling and culinary purposes, steam whistle, &c.

CHAPTER VII.

FITTINGS AND MOUNTINGS OF BOILERS.

Funnel and uptake.—The smoke and hot gases after leaving the smoke-boxes are conveyed from the different boilers, through passages called the uptakes, to the funnel, a sketch of which is given in Fig. 36. The area of the funnel in marine boilers is usually from one-seventh to one-tenth the area of the fire-grate.

The plates separating the several uptakes should, as pointed out at p. 80, be carried a sufficient height up the funnel to cause the smoke and gases to be moving in approximately the same direction when they mingle together, so as to avoid loss from shock and confusion of currents.

The funnel is surrounded at its lower part by an outer casing forming an annular air space. The heat radiated from the funnel is thus intercepted, and the ascending current of air is carried up into the atmosphere a sufficient height above the upper deck to prevent inconvenience to the people on board.

Fig. 36 shows what is generally known as a telescopic funnel, the upper part being so fitted that when the ship is under sail or in harbour it may be lowered by suitable apparatus, so as not to be much above the topsides of the ship. The funnel is only raised to its proper height when the ship is under steam.

Many of the masted ships of the Royal Navy are fitted

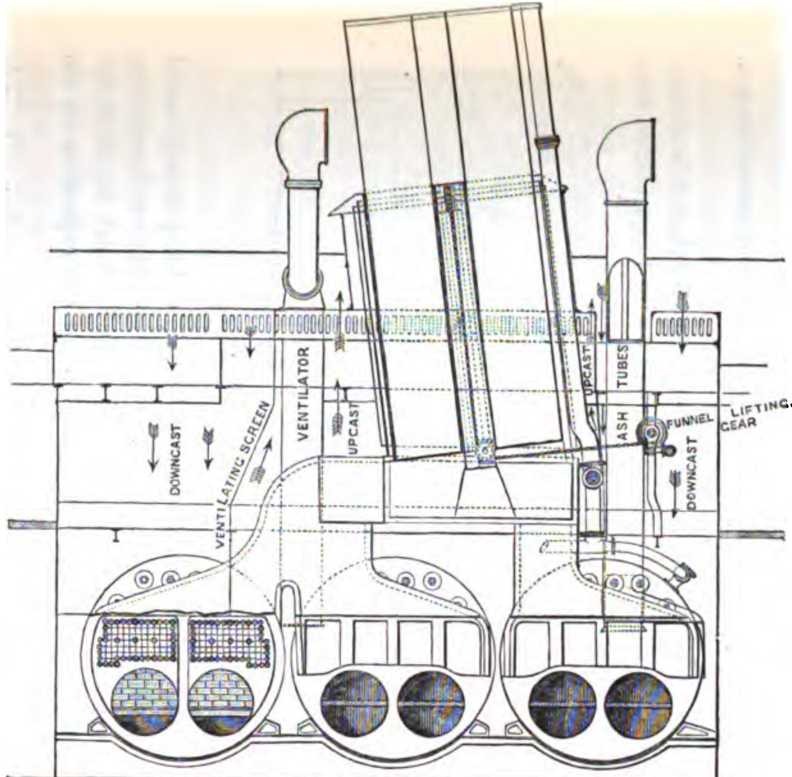

DOWNCAST
VENTILATING SCREEN
VENTILATOR
UPCAST
UPCAST
ASH TUBES
FUNNEL LIFTING GEAR
DOWNCAST

Fig. 36.

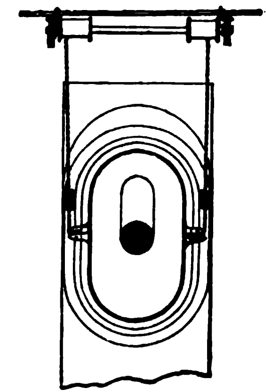


Fig. 37.

in this way, but though it may be a convenience in sailing ships, it reduces the evaporative power of the boilers, in consequence of the cold air entering the funnel through the annular space between its sliding and standing parts and checking the draught. In troop ships and turret ships in which sail power is of little or no moment, the funnels are generally fixed and not telescopic.

Funnel-lifting gear.—The gear for raising and lowering the funnel usually consists of chains or steel wire ropes passing over pulleys and wound up on barrels of small diameter, the power being obtained by worms and worm-wheels, as shown in Figs. 36 and 37, the latter being a plan of the gear.

Funnel stays.—When the funnel is raised it is secured by keys, and stayed by chains or wire ropes carried from

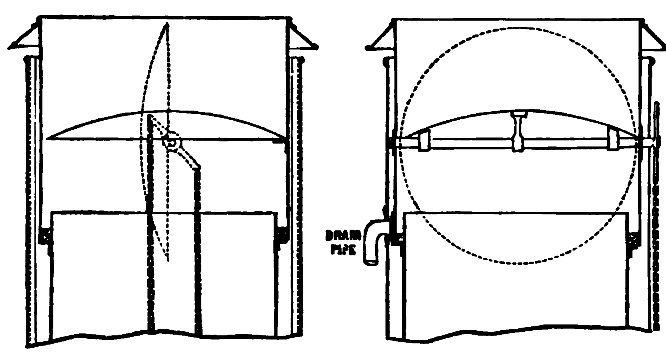


FIG. 38.

the top of the funnel to the ship's sides, usually called the funnel stays or guys. These are fitted with adjusting screws to regulate the strains, and should be slackened before raising steam to allow for the expansion of the funnel as it becomes heated.

Funnel cover.—When the ship is under sail, or in harbour, portable covers are fitted on the top of the funnel

to prevent rain water coming down and corroding the up-takes, &c. The covers are kept a little above the top, so as to allow sufficient space for the escape of the smoke from the small fires used for airing and warming the boilers. In many modern ships with fixed funnels, hinged dampers, worked from the deck, are fitted. A sketch of this is given in Fig. 38. In this case suitable drain pipes are fitted to carry off the rain water.

Ventilation.—The ventilation of the stokehold, when natural draught only is used, is obtained by a suitable arrangement of screens to separate the downcast from the upcast currents of air. The principal point to be kept in view in all schemes of ventilation is the provision of definite courses for the various columns of air to follow, so that there may be no confusion of currents, and that the streams of air, attempting to move in different directions, may not destroy each other and render the circulation stagnant. This would be the case if no divisional plates were fitted.

In the ventilation of a stokehold there are two points to be considered: first, the supply of a sufficient quantity of air for the fires; and secondly, the removal of the hot air and provision of fresh air for respiration, and for the reduction of the temperature of the stokehold, in order that the men may work as comfortably as possible. The general plan by which this is accomplished is shown in Fig. 36. Sheet-iron screens are fitted so as to form down-cast shafts to convey the cold air, from suitable hatchways, to each end of the stokehold. The greater portion of this air passes through the fires, and is used for the combustion of the coal. The central part of the stokehold is arranged to act as the upcast shaft, and the hot air from the fronts and tops of the boilers passes up this shaft and escapes through a casing round the funnel on the upper deck, its place being taken by a portion of the fresh air that enters

the stokehold by the downcast shafts, thus providing fresh air for respiration, and for keeping the temperature of the stokehold within reasonable limits.

Whenever possible the downcast passages should be cased in between decks to increase the height and consequently the weight, of the descending columns of air, and to insure fresh air being admitted to the stokehold, and not vitiated air from the lower deck.

In the smaller classes of vessels, in which the depth of hold is not sufficient to insure good ventilation by means

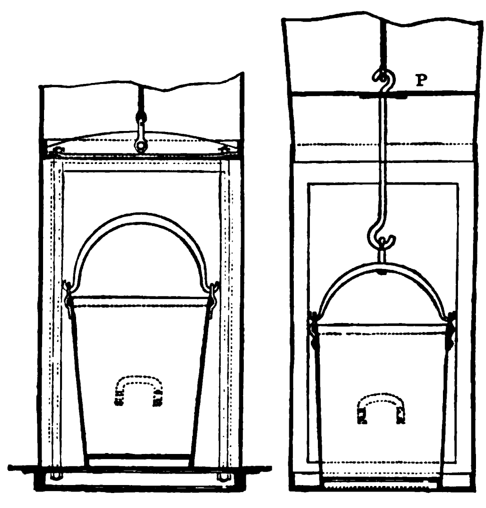


FIG. 39.

of the screens alone, circular pipes or tubes from the upper deck, with cowl-heads above the hammock-nettings, which can be turned in any direction to suit the prevailing direction of the wind, are fitted to increase the supply of air to the stokeholds.

In all new ships for the Royal Navy closed stokeholds and blowing fans are fitted to enable the boilers to be

worked with forced draught. The details of this arrangement are described in Chap. V.

Ash-shoot.—In all vessels there is at least one tube fitted for the purpose of raising ashes from the stokehold to the deck to be thrown overboard. These ash-tubes or shoots are utilised as ventilators. The lower ends of the ash-shoots are often made telescopic, so that they may be lowered to within four feet from the stokehold plates, to prevent accident. In closed stokeholds the ash-tubes have to be specially fitted to prevent loss of air during the process of raising ashes, when working under forced draught. Two methods of effecting this are shown in Fig. 39. In Fig. 39 A the ash-buckets are carried up and down in a frame or cage which works practically airtight in the tube. Fig. 39 B shows a more simple plan which is equally efficient.

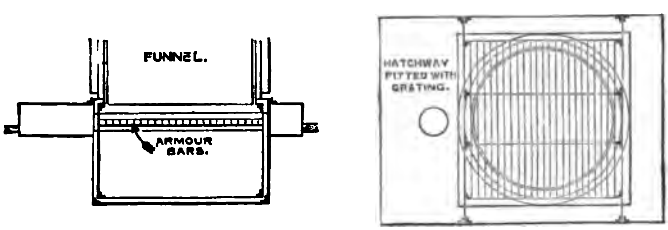


FIG. 40.

An iron plate, P, of sufficient weight to resist the air-pressure in the stokehold is carried loosely on the bucket chain, and acts as a valve when the bucket is at the bottom of the shoot, enabling the door to be opened and buckets to be taken out and put in as required.

Armour gratings.—In armour-clad and many other recent ships the hatchway round the bottom of the funnel, and the throat of the funnel itself, are fitted with deep iron bars or gratings to protect the uptakes and boilers from danger from shot or shell. The arrangement of the armour bars at the base of the funnel is shown in Fig. 40.

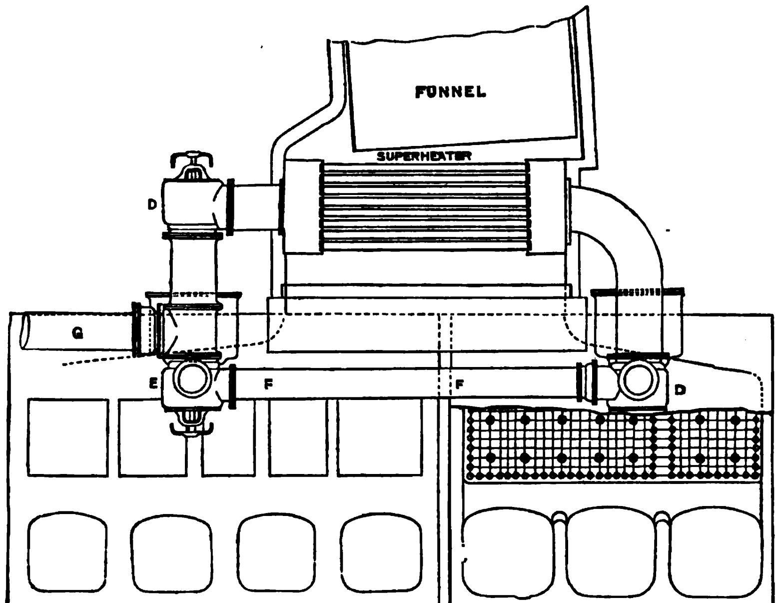


FIG. 41.

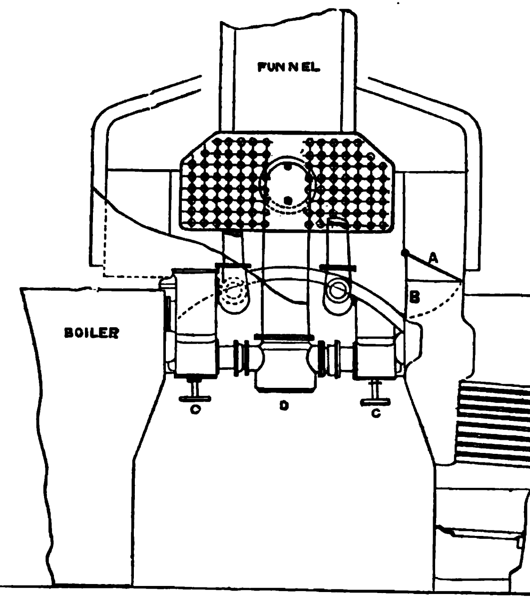


FIG. 42.

It will be seen that the plates at the sides of the funnel are carried across and secured to the ship by angle irons, so that the uptakes are relieved of the weight of the funnel and armour gratings, which are supported by the sides of the hatchway.

Superheater.—The general construction and arrangement of a superheater are illustrated by Figs. 41 and 42. It consists generally of a box of tubes fitted in the uptake; and the steam may be superheated by causing it to pass through these tubes before entering the main steam-pipe, the hot gases on their way to the funnel filling the space surrounding the tubes. Dampers are fitted in the uptakes, which may be placed either in the positions A or B, to compel the gases either to pass through the superheater tubes or direct to the funnel, and the superheater valves are so arranged that the steam may be made to pass either through the superheaters or direct to the engines, as may be required. In the sketch, C, C are the stop-valves on the boilers, and D, D and E the valves in connection with the superheater.

When it is desired to superheat the steam, the dampers are placed in the position marked A, Fig. 42, to cause the hot gases to pass through the superheater to the funnel; and the valve E is closed and D, D opened, so that the steam has to pass through the pipe F and the superheater to the main steam-pipe G which leads to the engines. When working with non-superheated steam the damper is placed in the position B, to allow the hot gases to pass direct to the funnel without going through the superheater; the valve E is opened and the valves D, D closed, so that the steam from the boilers may flow direct to the main steam pipe G without passing through the superheater tubes. As a general rule the three valves D, D, and E would in ordinary working be all kept partially open, so that the steam passing to the engines would be partly superheated and

partly saturated, and its temperature and dryness could be regulated, by suitable adjustment of these valves, to any degree desired.

Safety-valves.—The object of the safety-valve is to prevent the steam-pressure in the boiler from exceeding the safe working pressure. If all exit from a boiler were closed, and heat continuously applied, the pressure would continue to increase until at length the boiler itself must

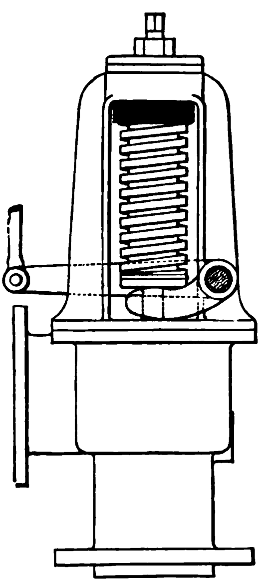


FIG. 43.

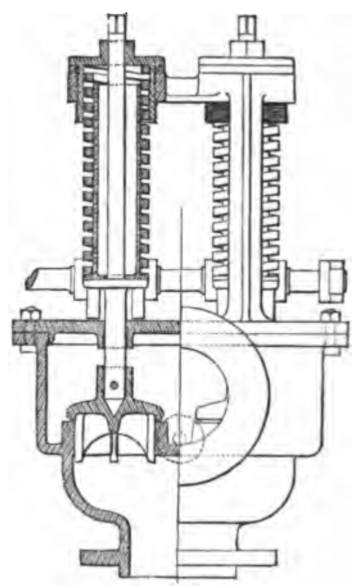


FIG. 44.

of necessity explode. The safety-valves are designed and fitted so that when the pressure in the boiler begins to exceed the safe working pressure, they open and let the excess steam pass off into the atmosphere, and thus prevent any dangerous increase of pressure occurring.

A general arrangement of safety-valves loaded with springs is shown in Figs. 43 and 44. The direct loaded spring valve has been chosen for illustration, as it is now

the most generally adopted for marine boilers. The general principles governing the design and arrangement of safety-valves are the same whatever their description, so that the explanation of this type will suffice for all.

The safety-valve boxes are fitted to suitable orifices at or near the top of the boiler, distinct from the stop-valve box or internal steam pipe, and the valves are kept on their seats by weights or springs of sufficient force to just resist the maximum working pressure of the boiler. When the steam pressure exceeds this, it opens the valves and escapes to the atmosphere through an orifice in the side or end of the box, to which the waste steam pipe is connected. The springs are generally placed outside the box to prevent their corrosion from the action of the steam.

Weighted valves were usually loaded by lead weights placed directly on the spindle above the valve. These valves, for marine boilers especially, had many disadvantages. The increased weight added to the boilers was not inconsiderable, and the oscillation of the weights due to the rolling of the ship caused the valves and seats to grind away and become leaky. The reduction of the direct load on the valve by the heeling of the ship caused waste of steam, and consequently of fuel, from the lifting of the valve; and the action of the weights, when the ship was inclined, tended to bend the spindle and thus to jam the valve and render it inoperative. They were also an element of danger when, as was often the case, they were bolted on to the fronts of the boilers, the heavy valve-boxes causing severe strains on the securing bolts and thin iron shell plates by the rolling of the ship and beating of the valves. They had, however, the advantage, that when they began to open they did not require any increase of pressure to open them still further, as is the case with spring-loaded valves.

The more a spring is compressed the greater is the

pressure required to compress it still further; and, within proper limits, the pressure required to compress a spring a certain distance increases directly as the amount of compression of the spring. This was the most difficult point to be overcome in the introduction of spring-loaded safety-valves. The difficulty has, however, been practically surmounted by employing springs of sufficient length and diameter to give flexibility; and spring-loaded safety-valves are now in general use for marine purposes. They are of considerably less weight, and more easily examined and adjusted than weighted valves, and the pressure on the valves is virtually the same whatever the inclination of the ship may be.

It appears to be undesirable to place the safety-valves in a horizontal position if it can be avoided. The guide feathers on the valve are always made with a certain amount of play, to prevent their sticking, and with horizontal valves, when the steam pressure is equal to the tension of the spring and the valve is in equilibrium, it drops from its central position and allows steam to leak past the seating. It is found necessary in such cases to increase the load on the valve above the required working pressure of the steam. Spring-loaded safety-valves, however, will work well in any position above or below the horizontal, the weight of the valve being allowed for in determining the initial compression of the spring.

The Board of Trade conditions under which spring-loaded safety-valves may be passed by their surveyors for passenger steamers are as follows:—

1. That at least two separate valves should be fitted to each boiler.

2. That the valves are of the proper size (see table in Appendix).

3. That the spring and valve be so cased in that they cannot be tampered with.

4. That provision be made to prevent the valve flying off in case of the spring breaking.

5. That the requisite safety-valve area be cased in, in the usual manner of Government valves.

6. That screw-lifting gear be provided to ease all the valves if necessary when the steam is up.

7. That the size of the steel of which the springs are made is to be found by the following formula:—

$$d = \sqrt[3]{\frac{W \times D}{C}};$$

where

W = load on the spring in pounds.

D = diameter of the spring, from centre to centre of the wire, in inches.

d = diameter, or side of square, of the spring steel in inches.

C = 8,000 for round steel.

= 11,000 for square steel.

If $d_{,}$ be the diameter of the steel in sixteenths of an inch, this rule becomes

$$d_{,} = \sqrt[3]{\frac{W \times D}{2}} \text{ for round steel;}$$

$$= \sqrt[3]{\frac{W \times D}{3}} \text{ for square steel.}$$

8. That the springs be protected from the steam and impurities issuing from the valves.

9. That when the valves are loaded by direct springs, the compressing screw is to abut against a metal stop or washer when the load sanctioned by the surveyor is on the valve.

10. Before the valve is passed the surveyor has to cause the boiler to be tried under full steam and full firing for

at least twenty minutes, with feed and stop-valves shut off, and in this case the accumulation of pressure must not exceed 10 per cent. of the loaded pressure.

The total area allowed for safety-valves in the Royal Navy has hitherto usually been half a square inch for each square foot of fire-grate surface; and for ordinary requirements this has been found to be ample. For the higher pressures this area may be reduced, because the rate of efflux of steam increases with the pressure, and the safety-valve area should depend on the pressure of steam as well as on the extent of grate surface. This is recognised in the Board of Trade rules, by which the proportionate area is reduced as the pressure is increased.

The area of the safety-valves should be based on the quantity of steam the boiler is capable of producing when worked at full power, and not on the grate area, because the rate of combustion varies considerably under different circumstances. This is rendered especially necessary by the introduction of closed stokeholds and forced draught, which has so largely increased the generative powers of boilers. The steam-producing power of the boilers may be represented approximately by the maximum I.H.P. developed, and the rate of efflux of the steam will vary as the absolute pressure. If P represent the absolute working pressure of steam, the total area of safety-valves required may be calculated from the formula

$$\text{area} = 3 \cdot \frac{\text{I.H.P.}}{\text{P.}}$$

This formula is obtained by taking the rate of flow of the steam through the orifice, in pounds per minute, to be $\frac{3}{4}$ths the absolute pressure in pounds, and assuming the valve to lift $\frac{1}{24}$th of its diameter when blowing off all the steam the boiler can produce if worked to the fullest extent of its power.

The face of a safety-valve should be somewhat lower than the body of the valve, as shown in the sketch, so that the pressure may be kept up when the valve lifts, by the action of the steam against the rim. Flat faces appear to be the most suitable for safety-valves, and the breadth of the face need not be more than from one-twelfth to one-eighth of an inch. With conical valves the seatings should be narrow, and fitted so as to be quite tight at the bottom of the cone; otherwise the actual area on which the steam-pressure acts will be greater than the nominal area of the valve, so that steam will be wasted, as the valves will lift before the steam attains the proper working pressure.

It has been proved by experiment that when the absolute pressure of the medium into which the steam enters on passing the orifice, is not more than one-half the initial absolute pressure of the steam, the velocity of issue is constant; that is to say, steam of 80 pounds absolute pressure would pass as freely into a chamber containing steam or air at 40 pounds absolute pressure as it would into a vacuum. In other words, the weight of steam that would pass through a given orifice per minute is constant when the outside absolute pressure is between zero and one-half the initial absolute pressure of the issuing steam.

It is desirable that the requisite safety-valve area for each boiler should be divided between two or more valves and not concentrated in one. The area for discharge of the steam is equal to the lift of the valve multiplied by its periphery, so that when two or more valves are used, the lift required to liberate the steam is reduced because the periphery is increased. The division of the safety-valve area has also the advantage of reducing the danger from the valves sticking fast, as the probabilities are against the contingency of all the valves becoming inoperative at the same time; and if one become set fast, the others

would still act to free the boiler of undue pressure if necessary.

In the Admiralty service all safety-valve springs, before they are passed, have to undergo the following practical tests as to their strength and elasticity.

The spring is first compressed by a load equal to the area of the valve, multiplied by the maximum working pressure per square inch of the steam in the boilers, and the extent of the compression is noted. The spring is then further depressed through a distance equal to one-quarter the diameter of the valve. This operation is mainly to test the strength and flexibility of the spring, and if on the removal of the load there be any permanent set, so that the spring does not regain its original length, it is rejected as not being sufficiently elastic. The load to produce the extreme compression should in no case exceed the maximum working load by more than 50 per cent. The present Admiralty practice is to make the amount of compression to give the working load not less than the diameter of the valve, so that the increase of pressure at the extreme compression would not exceed the working load by more than 25 per cent.

When the spring is placed on the valve a metal washer or ferrule is placed under the compressing screw, so that the spring cannot be compressed more than sufficient to give the maximum working load on the boilers. Before the safety-valve boxes are closed up, the valves are taken out and carefully measured and examined, to see that they are sufficiently free on their seats. For a 3-inch valve the clearance of the guide feathers should be about one-sixteenth of an inch.

The following results of the tests of some spiral safety-valve springs will illustrate the method of conducting the operations and the observations noted.

Diam. and section of wire	Diam. of helix from centre to centre of wire	Pitch of spring	No. of coils including two ends	Length of spring uncompressed	Diam. of safety-valve	Pressure per sq. inch of steam in boiler	Load applied for working pressure	Depression	Load required to open safety-valve ¼ diam.	Pressure per sq. inch required to open valve ¼ diam.
ins.	ins.	ins.		ins.	ins.	lbs.	lbs.	ins.	lbs.	lbs.
$\frac{7}{16}$ round	$2\frac{3}{8}$	$\frac{15}{16}$	11	$10\frac{3}{4}$	$2\frac{11}{16}$	60	341	$1\frac{5}{16}$	525	$92\frac{1}{2}$
$\frac{7}{16}$,,	$3\frac{1}{16}$	$1\frac{3}{16}$	11	$12\frac{7}{16}$	$2\frac{1}{2}$	60	294	$2\frac{3}{8}$	374	$76\frac{1}{2}$
$\frac{7}{16}$ square	$2\frac{11}{16}$	$\frac{11}{16}$	17	$12\frac{9}{16}$	$2\frac{1}{16}$	60	341	$1\frac{1}{2}$	491	$86\frac{1}{2}$
$\frac{3}{8}$ round	$3\frac{1}{2}$	1	18	$17\frac{7}{8}$	$3\frac{3}{4}$	30	331	$2\frac{13}{16}$	439	$39\frac{3}{4}$
$\frac{1}{2}$,,	3	$1\frac{1}{8}$	10	$10\frac{7}{8}$	$2\frac{3}{4}$	60	357	$1\frac{7}{8}$	514	$86\frac{1}{2}$
$\frac{5}{8}$,,	$2\frac{1}{2}$	1	13	$12\frac{1}{2}$	$3\frac{5}{8}$	60	620	$1\frac{1}{2}$	1015	$98\frac{1}{4}$
$\frac{1}{2}$,,	3	$\frac{3}{4}$	18	14	3	60	425	$2\frac{7}{16}$	561	$79\frac{1}{4}$
$\frac{1}{2}$ square	$2\frac{1}{2}$	$\frac{7}{8}$	14	$12\frac{5}{8}$	$3\frac{5}{8}$	60	619	$1\frac{3}{8}$	1116	$108\frac{1}{4}$
$\frac{9}{16}$ round	$5\frac{11}{16}$	$1\frac{9}{16}$	13	21	$4\frac{3}{4}$	60	1063	$6\frac{3}{8}$	1246	$70\frac{1}{4}$
$\frac{5}{8}$,,	4	$1\frac{1}{8}$	16	$17\frac{7}{8}$	$5\frac{1}{4}$	30	648	3	938	$43\frac{1}{4}$
$\frac{5}{8}$,,	4	$1\frac{3}{8}$	9	$12\frac{1}{16}$	$3\frac{1}{2}$	60	578	$1\frac{1}{2}$	931	96
$\frac{5}{8}$,,	$3\frac{1}{2}$	1	8	$8\frac{7}{8}$	$2\frac{3}{4}$	120	713	$1\frac{3}{16}$	1146	193
$\frac{3}{4}$,,	$7\frac{7}{8}$	$2\frac{1}{32}$	10	21	$4\frac{3}{4}$	60	1063	$6\frac{3}{8}$	1246	$70\frac{1}{4}$

Safety-valve lifting gear.—The gear used for lifting the safety-valves is shown in Figs. 43 and 44. It usually consists of levers acting either under the valves themselves or under collars on the valve spindles, these levers being worked by screw gear on the fronts of the boilers or in some other convenient place. It is now the general practice in the Royal Navy to fit gear so that the valves may be lifted from the deck as well as from the stokehold, the two sets of gear working independently of each other. All the joints in the safety-valve easing gear should either be fitted with gun-metal bushes, or the joint pins should be of gun-metal to prevent the danger of the gear rusting and setting fast. Care must be taken that the gear shall not in any way impede the lifting of the valve when acted on by the steam pressure in the boiler. The sole object of the lifting gear is to provide the means for raising the valve by hand when necessary; and it must not be so fitted as to keep the valve on its seat, which must be done by the action of the springs or weights alone.

The safety-valve box should always be bolted directly on the boiler, so that no other valve nor any obstacle can possibly intervene to prevent the steam pressure from always acting directly on the safety-valves themselves.

Pressure gauges.—The steam pressure in the boiler at any time, or, more strictly speaking, the excess of the pressure in the boiler above that of the atmosphere, is usually indicated by Bourdon's pressure gauges, or other gauges of somewhat similar construction.

Fig. 45 shows their general construction and arrange-

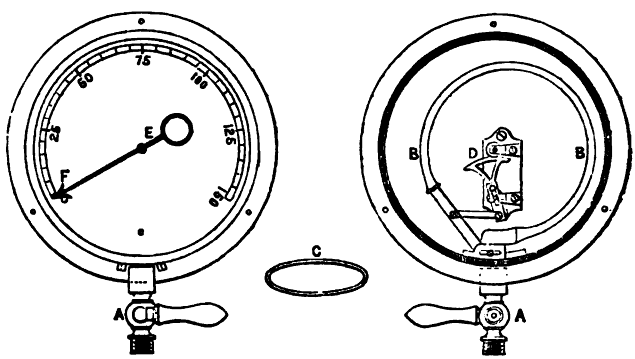


FIG. 45.

ment. A is a cock on a small pipe connected to the boiler. BB is a curved metallic tube, of elliptical section, as shown to enlarged scale at C, which is closed at one end and open to the steam pressure through the cock A at the other. The greatest breadth of the section of the tube is perpendicular to the plane in which the tube is curved. The closed end of the tube is connected by a sector D to a small pinion on the axis of the index finger EF, which points to a graduated arc. When the pressure inside the tube is greater than the external pressure, the tube becomes less curved, and this causes the sector to act on the

pointer and indicate the pressure. The lower end of the sector lever D is in two parts, connected by a tightening screw; the hole in one part is elongated to allow the instrument to be adjusted by altering the length of the lever. The free end of the elliptical tube BB is fitted with a stud, working in a slot in the piece that secures the tube to the bottom of the gauge casing; this is to prevent the possibility of the sector being carried over far enough to be thrown out of gear with the pinion on the index finger.

The graduation of these gauges for the limits of pressure within which boilers are usually worked is obtained by comparison with a mercurial gauge into which mercury can be pumped to sufficient heights to obtain the necessary graduations on the Bourdon gauge.

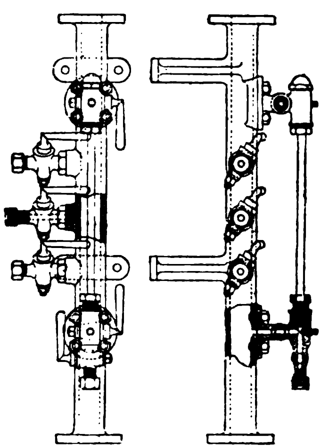


FIG. 46.

In the Royal Navy each boiler is fitted with two pressure gauges, in order to provide for the case of one gauge getting out of order, and to be a check on each other. One of these gauges is graduated to rather more than double the working pressure of steam, to enable it to be used for the purpose of testing the boiler by hydraulic pressure to twice the working pressure at regular stated intervals.

Water gauges.—The level of the water in the boiler is indicated by a glass tube fitted between two cocks, one in connection with the steam chest and the other with the water space of the boiler. The general arrangement is

shown in Fig. 46. In most cases these cocks are fitted on a brass pipe, sometimes known as a *steady* or *standpipe*, the top of which is connected to the steam chest and the bottom by an outside pipe to the lower part of the boiler. Sometimes, however, the gauge cocks are bolted direct to the fronts of the boilers themselves. The fitting which forms the lower branch of the glass water gauge arrangement consists of two cocks, one communicating with the boiler, and the other forming a drainpipe for blowing through the apparatus and keeping it clear. A small screw is fitted in each cock opposite the holes, to enable a wire to be passed in to clear them when necessary. Two sets of water gauges are fitted to each boiler in ships of the Royal Navy.

The general practice is to fix the hole in the lower gauge cock at the level of the highest part of the heating surface so that, in consequence of the space taken up by the gland, &c., the water, when the level is reduced to the bottom of the glass and is just getting out of sight, is from two to three inches above the highest part of the heating surface. The total length of the gauge-glasses used in H.M. service is sixteen inches, external diameter $\frac{5}{8}$-inch, and thickness $\frac{1}{8}$-inch. Allowing for the part of the length of the glass tube in the glands of the cocks, the depth of the water over the highest part of the heating surface when the glass is half-full, will be found to be about nine inches.

In some cases, in which the lower cock of the glass gauge has been bolted directly to the boiler, at or near the level of the highest part of the heating surface, it has been connected to the lower part of the boiler by a pipe inside. This practice is incorrect and dangerous, as by opening the drain cock on the lower part of the gauge apparatus, water would issue, even if the level were considerably below the top of the heating surface, which might cause

misleading deductions to be drawn as to the safety of the boiler. If the internal pipe be carried up open-ended into the steam space as well as down into the lower part of the boiler, the indications will be reliable, but in general it is better to dispense with internal pipes altogether.

Test cocks.—In the centre part of the steady pipe, or when this is not fitted, in the front of the boiler itself, two, sometimes three, small cocks, called test or gauge cocks, are usually fitted. The orifice of the lower of these cocks is about two or three inches above the highest part of the heating surface, and that of the upper cock a little above the normal water-line. When a third cock is fitted it is placed midway between the other two, but this cock is not necessary, and is therefore but seldom fitted. The use of these cocks is to enable the level of the water to be ascertained approximately in case of accident to the glass gauge. When working by these cocks it is clear that if, on opening the upper cock, steam only issues, the water is not too high in the boiler; and as long as water can be drawn from the lower cock we may feel sure that the water-level is not dangerously low, as the heating surface will be covered with water. The water from the lower cock will partially evaporate on being drawn off, as its temperature will be above the boiling point of water at atmospheric pressure. These test cocks should be fitted without internal pipes.

Feed valves.—These are the valves through which the feed water is admitted to the boilers. They are generally simple non-return conical valves, with regulating or set screws above them, by which they may be kept closed, or the amount of lift regulated, as may be necessary. These valves are made non-return, so that in case of the feed pumps ceasing action from accident or other cause, the water in the boiler may not be forced back through the

pipes by the steam pressure. They also prevent the boiling water from acting on the valves of the feed pumps when the engines are stopped, and enable them to be examined if necessary. An additional non-return valve is usually fitted in the delivery pipe of each feed pump to facilitate this operation.

Two of these valves are fitted to each boiler, one, called the main feed valve, being in connection with the main engine feed pumps, and the other, called the auxiliary or donkey feed valve, connected with the auxiliary or donkey feed pumps.

Blow-out cocks.—These cocks are fitted to the bottoms of all marine boilers, and are used for the purpose of filling and emptying the boilers, and, during the working of the engines, for blowing out part of the water from the bottom of the boiler when necessary, so that the water may be changed and prevented from becoming too salt or impure. When the blow-out cocks are used, the feed valves should be shut off to insure the impure water being expelled.

In boilers that are fed entirely with sea-water from jet condensers, it is necessary that these cocks should be used regularly to prevent the density of the water in the boilers from becoming too high. In ordinary practice in boilers of this class the water is kept at a density ranging from $2\frac{1}{2}$ to 3 times that of sea-water, according to circumstances. When any particular density has been selected as the most suitable for the conditions under which the boilers are worked, it should be kept constant to prevent waste from irregular or excessive blowing off. The higher the density at which the water in the boiler is kept the smaller is the quantity of water that is required to be blown off, and consequently a less quantity of heat is wasted. The proof of this will be found in Chapter XII.

The blow-out cock is generally an ordinary straightway cock, and is connected either directly to the bottom

of the boiler or to a strong gun-metal pipe bolted to the bottom. To avoid accident from leaving the blow-out cock open, a guard is fitted to prevent the spanner from being removed without first closing the cock; so that when the spanner is off, it is certain that the cock must be shut. In some cases the guard is fitted to the sea cock of the blow-out Kingston valve common to several boilers. In these cases separate blow-out cocks or valves are fitted to each boiler, so that either may be blown out independently of the others.

Hydrometer.—The density of the water in the boiler is ascertained by means of an ordinary hydrometer. This is an instrument of the form shown in Fig. 47, made either of glass or metal. It has a slender stem A, and two bulbs, the larger one filled with air to give buoyancy, and the smaller one, C, loaded, to keep the instrument vertical when placed in a liquid.

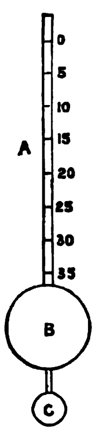


Fig. 47.

When any body floats in a liquid, the weight of the liquid displaced is equal to the weight of the body, so that the higher the density of the liquid the less depth will the body sink in it.

The vertical stem of the hydrometer used in the Navy is graduated in the following way. The zero point is marked at the level at which the instrument floats in fresh water. It is then immersed in sea-water of average density, and the level of flotation again marked. The space between these two points is divided into ten parts, which are called the degrees of the hydrometer, and the graduations are carried up to about 45°, which is sufficiently high for the purpose.

Ten degrees, therefore, represent the density of sea-water, or $\frac{1}{33}$ part of solid matter, and every additional ten degrees represent an increase of density due to an

additional $\frac{1}{33}$ part of solid matter. The density of the water is generally given in accordance with this scale:

10°	representing density of sea-water.				
15°	„	$1\frac{1}{2}$ times density of sea-water.			
20°	„	2	„	„	„
30°	„	3	„	„	„

and so on.

As the density of water depends on its temperature, a thermometer is required as well as a hydrometer, in order to determine exactly the density, and in the salinometer, which was at one time used, the two instruments were combined. Practically, however, this difficulty has been overcome by graduating the hydrometers to suit a uniform temperature of 200° Fahr., which is about the temperature the water would be a few seconds after being drawn off into the hydrometer pot for testing. This plan has been found to be sufficiently accurate for the purpose, and is now generally adopted, as it avoids the complication involved in the use of two instruments.

Surface blow-out or brine cock.—In most marine boilers, cocks or valves are fitted to draw off the scum and impurities floating on the surface of the water. These are known as surface blow-out, brine or scum cocks or valves. From this cock or valve, which is fitted to any convenient part of the boiler, a pipe is carried inside to the central part of the boiler, where it terminates in an open pan placed a little below the water-level. When the cock or valve is opened a vortex is formed above the pan into which the grease and impurities are carried and are discharged overboard.

Stop or communication valves.—These valves are for the purpose of regulating the passage of the steam from the boilers to the engines. One of these valves is fitted to each boiler, and is connected to the main steam pipe, so

that any, or all of the boilers, may be placed in communication with the engines as may be required. Its general form and construction are shown in Fig. 48. It is fixed so that the pressure of the steam in the boiler may be inside the valve, which is worked by means of a screw, the spindle passing to the outside of the valve box through a steam-tight stuffing box and gland as shown.

Self-closing stop-valve.—In order to lessen the damage resulting from explosion, or other accident to a boiler, as, for example, its being pierced by a shot, stop-valves in war-ships are now always made self-closing. This

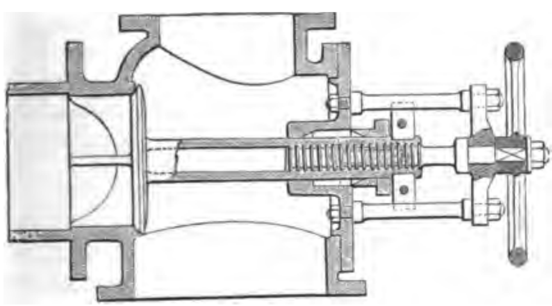


FIG. 48.

form of valve is shown in Fig. 49. It will be seen that the valve is simply a non-return valve, the action of the screw outside being only to keep the valve closed on its seat, or to regulate the amount of its opening. When the screw is worked back the pressure of steam in the boiler, if it be greater than that in the steam pipes, will force the valve open so as to allow the steam to pass to the engines. In the event of the pressure in any boiler falling suddenly from any cause, the pressure in the steam pipes would close the valve and isolate the boiler.

With the ordinary stop-valve, if a hole were made in

any one boiler, the whole of the boilers would be rendered useless, until the stop-valve of the injured boiler could be closed by hand, which, in all probability, would not be until the steam from all the boilers had discharged itself through the damaged boiler, increasing the extent of the disaster, and rendering the ship for the time perfectly helpless.

It will, therefore, be seen how valuable these valves

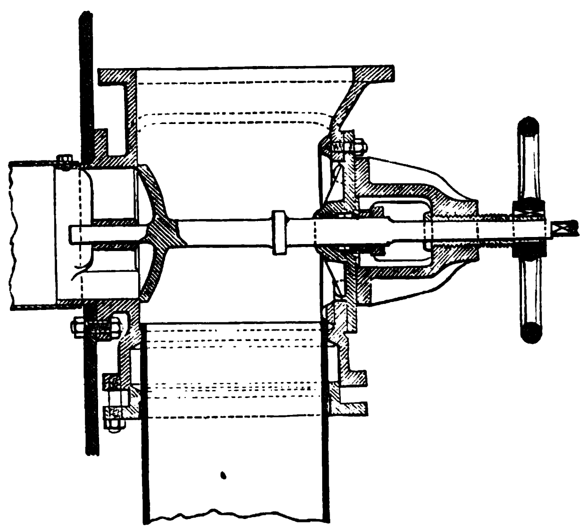


FIG. 49.

would probably be in a war-ship in case of action. In the event of a shot entering a boiler, the damage would be localised, being limited to the boiler struck, and only one boilerful of steam would be admitted to the ship instead of the steam from the whole of the boilers, which would most probably be the case with ordinary stop-valves.

Valves of this description should always be placed in or

below a horizontal position, so that the weight of the valve should have no effect in closing it. When placed vertically, the pulsations of the steam in the steam-pipe cause the valves to work up and down on their seatings with such violence as to threaten to break either the valves themselves or the valve-boxes. In some cases the valves have been broken to pieces from this action.

Internal steam-pipes.—Stop-valves have attached to them, inside the boiler, thin brass pipes, which are carried across the steam chest close to the top of the boiler, and are called the internal steam-pipes. They are closed at the end, but have narrow slits cut at the top, which allow the steam to enter the pipe and pass through the stop-valve to the engines. The object of this fitting is to prevent as far as possible priming, or the passage of water through the stop-valves with the steam. Before they were fitted it was found that in some boilers, on the rolling of the ship, large quantities of water entered the steam-pipes and were carried to the engines, producing often injurious results. In any case there is little doubt that by spreading the area of collection of steam over the boiler the evaporation is rendered more uniform, and the tendency to priming from the rush of the steam to a single orifice is obviated. Every care should be taken to prevent the slits from being choked, especially in boilers that have a tendency to prime.

From the stop-valves on the different boilers steam-pipes are carried, which unite at the end of the stokehold in one main steam-pipe through which the steam passes to the engines.

Bulkhead stop-valves.—In ships that have more than one stokehold, the steam-pipe from each stokehold is carried independently to the engine-room bulkhead, and at the end of each pipe a bulkhead self-closing valve is fitted, so that in the event of any steam-pipe being damaged only

one set of boilers would be put out of work, the others remaining efficient. The steam-pipes from the several boiler-rooms are connected by a cross pipe in the engine room to enable either or both sets of engines to be worked from any boiler, and screw-down valves, which can be worked both from the engine rooms and the deck, are fitted between the bulkhead valves and the regulating valves for each set of engines.

Separator.—Before reaching the engines the steam has to pass through the separator, a sketch of which is given in Fig. 50. The object of this fitting is to provide an additional safeguard against priming, by preventing any water that may have passed the stop-valves and got into the steam pipes from entering the cylinders. It is divided from the top to nearly the bottom by a diaphragm, D, the steam entering on one side and leaving on the other. Any water that reached the separator would thus be left at the bottom, the steam only, rising to the exit orifice and passing on to the cylinders. In machinery with jet injection condensers the water from the separators is got rid of by being blown overboard through the drain cock, but for surface-condensing engines an additional pipe with non-return valve is fitted, so that the water may be discharged into the hot well or feed tank if desired, and returned to the boiler as feed, instead of being wasted and necessitating the admission of sea-water to supply its loss.

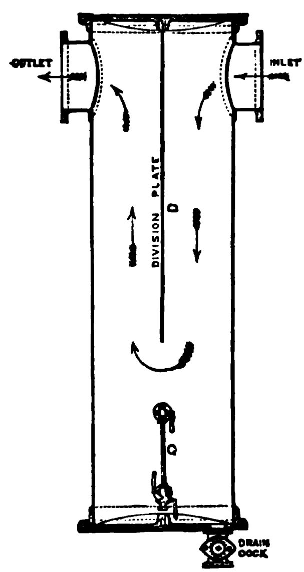


FIG. 50.

The level of the water in the separator at any time is shown by a glass gauge, G, similar to that fitted on the boilers.

Auxiliary steam service.—Many marine boilers, in addition to the main stop-valves and steam-pipes, are fitted with an auxiliary steam service consisting of small stop-valves and steam pipes, similar in construction and arrangement to the main valves and pipes, which may be used for the auxiliary engines, distilling purposes and for other similar uses. These fittings prevent the necessity of filling the whole range of steam-pipes with steam when one boiler is in use for distilling or other auxiliary purposes, the main valves and pipes only being used for the actual working of the engines.

For the higher steam pressures that are now becoming universal it is desirable, either to supply a special auxiliary boiler for distilling, or to fit an additional service of small pipes and valves to be used for this purpose only. As a rule, the steam pressure used for distilling is much lower than the ordinary working pressure; and if the general auxiliary steam service supplies the distilling condensers, the auxiliary stop-valves on all the boilers, except the one in use for distilling, must be kept closed, which might cause inconvenience if the feed-donkey or other auxiliary machinery, or the steam whistle, were required to be worked.

The special auxiliary service of steam valves and pipes has been dispensed with in the more recent ships, branches being led from the main steam pipes to supply the auxiliary engines. This arrangement is sufficient to meet all the requirements of the ship, and reduces complexity, especially when the pipes have to pass through several water-tight bulkheads, and also lessens the number of holes required to be cut in the shells of the boilers. In this case an additional stop-valve is fitted in the main steam-pipe, to

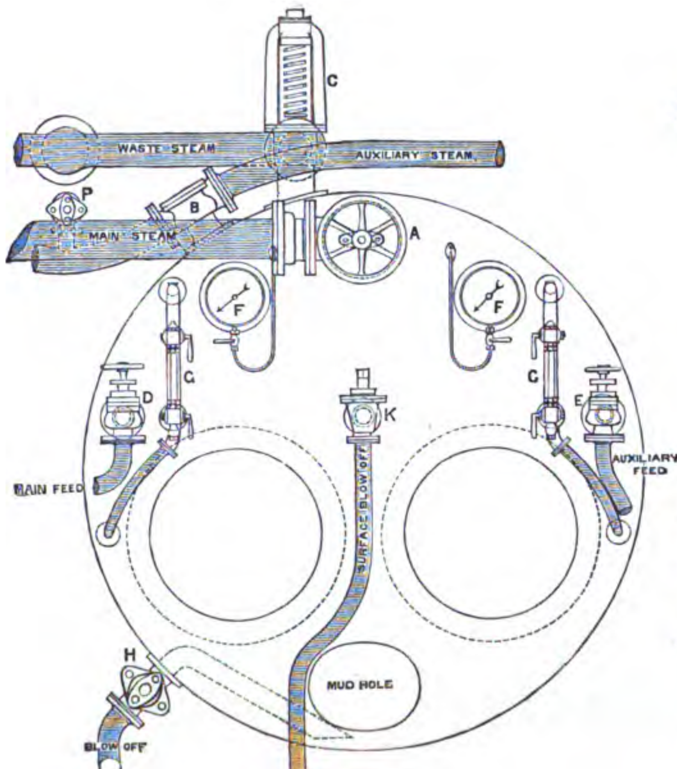
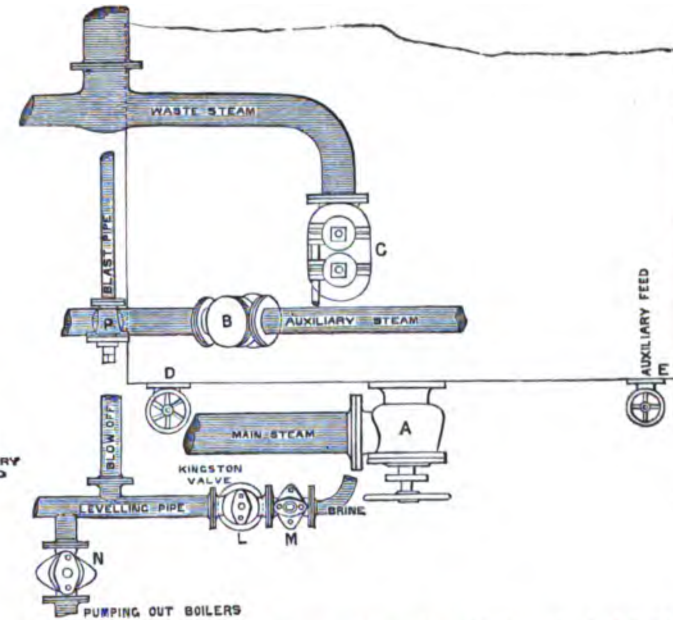


Fig. 51.



A is the main stop-valve. B is the auxiliary stop-valve. C is the spring loaded safety-valve. D is the main feed-valve. E is the auxiliary feed-valve. FF are the pressure gauges. GG are the water gauges. H is the blow-out cock with internal pipe leading to bottom of boiler. K is the surface blow-out or brine cock. L is the blow-out Kingston valve for both surface and bottom blow-outs. M is the sea-cock for the brine pipes. N is a cock leading to the hand pump to enable either of the boilers to be pumped out when necessary, through the levelling pipe and blow-out cocks. P is the blast cock, to admit a jet of steam into the funnel when required to quicken the draught through the fires.

Fig. 52.

keep the steam pressure off from the main engines when required for auxiliary purposes only. Separate pipes are taken direct from two or more boilers to the distilling condensers.

Figs. 51 and 52 show a general arrangement of the principal mountings on a cylindrical marine boiler. The joints between the mountings and the boiler should be perfectly tight, as any leakage would cause rapid decay of the shell of the boiler. All valves and cocks in connec-

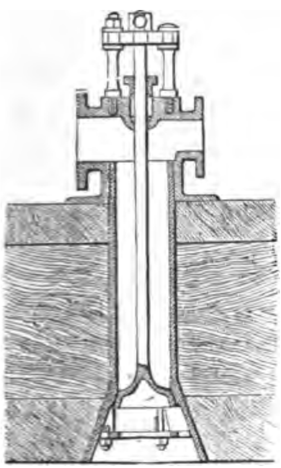


Fig. 53.

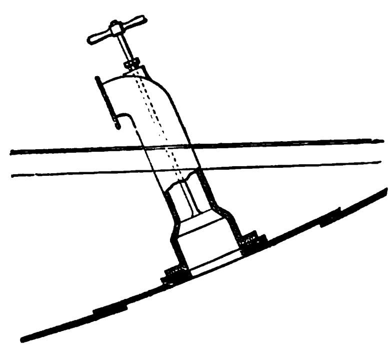


Fig. 54.

tion with the boilers are made with spigots passing through the plates to prevent corrosion.

Kingston's valves.—Holes in the hull of a ship below the water-line are generally closed by means of Kingston's valves, sketches of which are shown in Figs. 53 and 54, with their methods of attachment to wood and iron ships respectively. They are simply conical valves, arranged so that the pressure of the water outside tends to keep them closed on their seatings. The valves are fitted with long spindles, which are brought inside the ship through tight

stuffing-boxes, to enable the valves to be worked from inboard. The spindles of all Kingston valves should be solid with the valves, and in the Royal Navy all these spindles have to pass a tensile test, equal to half-a-ton per square inch of area of the valve; with this limit, however, that the maximum test load is not to exceed twelve tons whatever may be the diameter of the valve.

In the case of iron and steel ships, care must be taken to prevent decay of the skin of the ship in the region of the under water valve orifices, especially those for the boiler blow-outs. A thick plate is riveted to the skin of the ship inside, and the valve casing is secured to this instead of to the plating of the hull. The casing has a spigot on its end, which fits tightly in the hole to cover the edge of the plate, and zinc ring protectors are also fitted. This is shown to an enlarged scale in Fig. 55.

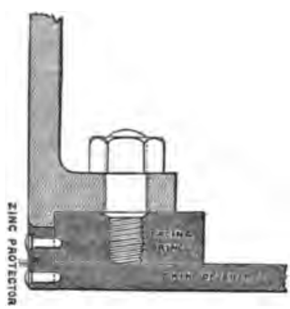


FIG. 55.

In the case of the boiler blow-out Kingston valves, it also appears to be necessary to fit covering plates outside the ship to protect the hull plating in the neighbourhood of the orifices. In ships that have double bottoms the Kingston valve casings are sometimes attached to the outer bottom, and brought through stuffing boxes on the inner skin; or iron or steel tubes are fitted between the two skins and the Kingston valves attached to the inner skin in the ordinary manner. All Kingston valves are fitted with sea cocks inside the ship for additional security.

These valves were particularly suitable for wood ships because they enabled a firm and secure connection to the hull to be conveniently made, as shown in Fig. 53. For

iron and steel hulls to which ordinary flange connections can be made, this form of valve has no special advantage, and in modern ships ordinary screw-down valves, which are cheaper and lighter, are generally attached to the openings in the hull instead of Kingston valves.

CHAPTER VIII.

CORROSION AND PRESERVATION OF BOILERS.

THE question of the durability of boilers is one of the greatest importance as regards the continued efficiency of steamships on active service, and much attention has consequently been paid to it, especially within the last few years. The evidence given before the Admiralty Committee on Boilers, by many of the leading engineers and chemists in the country, was of great value in clearing the ground, as from the contradictory statements made, it was evident that at that period all were pretty much in the dark on the subject, and that the question could only be solved by extended experiment and careful observation. The great variety of opinions and practice relative to this subject may be readily seen by reference to the analysis of evidence given in Appendix A, p. 118, of the Committee's Report as given in the 'Blue Book,' published in 1877. The researches and experiments of the Boiler Committee, which are published in their Reports of 1877, and the further experiments that the Admiralty have since that time carried out, have thrown considerable light on very many points, and the methods of treatment that have consequently been adopted have very materially increased the durability of boilers.

It is not our present intention to enter fully into the general question of the corrosion and preservation of boilers,

which would occupy more time and space than we have at command, but merely to skim lightly over the surface, and briefly indicate the causes that appear to affect decay, and the methods that should be adopted to neutralise them or reduce them to a minimum so far as our present knowledge and experience will guide us.

The question is one specially important to officers in command; and we trust this brief sketch of the probable causes of decay, with the remedies suggested, may enable the leading principles to be clearly understood, so that the captain and the engineer officer may be able to work together to attain the desired end. Necessarily the greater part of the work connected with the care and preservation of the boilers is professional, and mainly, if not entirely, lies in the province of the engineer; but many points, particularly those relating to the management of the boilers when on active service, are to a very great extent under the control of the captain, and the efficiency and durability of the boilers will depend greatly on his knowledge of the subject.

The efficiency of a war-ship in the present day may be measured mainly, if not wholly, by the efficiency of her machinery, so that if by mismanagement or want of care the boilers are injured the evil effect cannot be estimated by the depreciation in the value of the boilers alone, which would be comparatively a small matter, but it must be measured by the fact that the efficiency of the ship for the purpose for which she is designed is decreased, which is a much more important consideration. We would, therefore, venture to commend this chapter specially to the study of officers in command, or who may in future be in command, of any of Her Majesty's ships.

We would in the first place point out that whilst it is important that by proper care and precaution the boilers

should be enabled to retain the original working pressures for as lengthened a period as possible, it is necessary to guard against exaggerated ideas of the loss of efficiency resulting from the reduction of the load on the safety-valves as the boilers become worn. Boilers, like everything else, will wear out, and when the plates and stays become worn, it is certainly much better to reduce the working pressure in the boilers, especially in war-ships, in order to prolong their existence, than to lay the ship up and fit new boilers as soon as the boilers become unfit for the original load.

The engines are generally designed to work with a considerable amount of expansion at full power, and within certain limits the initial pressure may be reduced without any very great loss of power or efficiency for ordinary work, the only effect being to increase the consumption of coal. The gain resulting from the extended durability of the boiler, however, greatly exceeds the increased expenditure of coal due to the less economical working of the engines, and the ship may be kept in an efficient state for probably years longer than would otherwise be the case.

As an illustration of this we might mention the 'Minotaur.' When first fitted out in 1867, with a steam pressure of 25 pounds per square inch, her full speed on the measured mile was 14·3 knots. The pressure was reduced from time to time as the boilers became weaker, till, in 1872, the load on the safety-valves was reduced to 12 pounds per square inch; but it was found on trial that the speed was 12·8 knots, or only 1·5 knots less than her maximum speed, though the working pressure was reduced to less than one-half its original amount.

The increase in the pressures of steam generally used had a most important effect in reducing the useful durability of boilers. By useful durability is meant their

power of being worked at or within a reasonable range of the pressure for which both the engines and boilers were designed. When the ordinary working pressures were from 5 to 8 pounds per square inch, the boilers could be patched and kept going for almost any length of time. As the pressures were increased in rectangular boilers, the extra strength necessary was obtained by additional staying, and not by increasing the thickness of the boiler plates themselves to any considerable extent. With these boilers, therefore, when the plates became much thinned by wear, it was found to be more economical to fit new boilers than to attempt to repair the old ones for the original pressure, or something near it, though they might have been worked for a long time at the lower pressures previously used.

The general introduction of surface condensation affected the durability of boilers in two distinct ways and brought the subject more prominently into notice. In the first place it rendered the use of high-pressure steam for marine engines practicable. This can, however, only be considered as indirectly due to the surface condenser. In many cases, however, a special corrosive action was found to take place in boilers supplied with water from surface condensers, particularly at first, when the subject was little understood, and the extraordinary cases of rapid decay that occurred in boilers of this class under various circumstances compelled the whole question to be carefully investigated in order to ascertain as far as possible the nature of the action and its remedy.

As the tubes in the surface condensers were generally made of copper, it was at first supposed that galvanic action would account for the decay of the boilers, either by the copper condenser tubes and the iron boiler plates forming a great battery, or by the tallow used for lubrication forming fatty acids on decomposition by the heat, and carrying

with it into the boilers particles of copper dissolved from the condenser tubes or feed pipes. The fact, however, that the condenser tubes and feed pipes themselves generally appeared to suffer little or no deterioration after considerable periods of work, effectually disposed of this hypothesis.

The general opinion of the chemists who examined this question was against the theory of galvanic action, and it was concluded that the main causes of the rapid decay of the earlier boilers fed with water from surface condensers were, that the fatty acids evolved from the tallow and vegetable oils used for internal lubrication, and which were carried from the condensers into the boilers by the feed water, acted directly as corrosive agents on the iron of the boiler plates and stays and destroyed them. This action would be accelerated and intensified with superheated steam, which was used in the majority of these boilers and engines.

The extraordinary decay of some of the boilers of ships belonging to the P. and O. Company, when surface condensers were introduced in 1862, was investigated by three eminent chemists, viz. Mr. Ogston and Professors Letheby and Hofman.

Mr. Ogston's analysis of the deposit found in the boilers gave no trace of copper, and his opinion was that the destruction was unquestionably due to the acids given off by the heated tallow acting directly on the boilers and dissolving the iron. The irregularity of the action was explained by the fact that clean surfaces would be attacked most readily, and that all parts covered with a thin film of oxide or scale would be protected. He suggested as a remedy that caustic lime or soda should be pnt into the boilers from time to time to neutralise the acids in the water.

Dr. Letheby found that all the samples of water sent to

him contained the volatile acids of fat and distinct traces of copper. He also found a notable quantity of copper in the specimens of deposit taken from the boilers. He was at first led to the conclusion that the principal, if not the only, cause of the mischief in the boilers was the presence of a minute quantity of copper in the condensed water supplied to them, and that this copper came from the tubes of the surface condensers. The remedies he proposed were :—

1. The doing away entirely with all copper or brass upon the surfaces where the water is condensed, either by substituting iron or tin tubes for the copper or brass tubes ordinarily used, or by covering the tubes with enamel or other coating.

2. The protection of the inner surfaces of the boilers with some impervious or insoluble coating, such as Portland cement.

After this, however, he made further experiments with tinned and electro-plated tubes to test the value of remedy No. 1, but found that the water condensed by the coated tubes still acted powerfully on iron, and this he attributed to the same source that Mr. Ogston assigned as the general cause of corrosion, viz., the presence of fatty acids, and to the circumstance that water free from saline matter, and in the presence of atmospheric air, quickly oxidises iron.

These later experiments therefore threw considerable doubt on the reasons before assigned by Professor Letheby, for they clearly proved that the destructive action still proceeded notwithstanding the absence of any trace of copper in the water.

To obviate this action, Dr. Letheby proposed the admixture of sea-water with the water in the boilers and the addition of slaked lime, say a bushel or so to every hundredweight of grease used in the engines, to neutralise the fatty acids in the water of the boilers.

Professor Hofman entered more fully into the subject than the other two gentlemen named, and his investigations extended over a period of nine months. He also found slight traces of copper, about 0·4 per cent., in the specimens of deposit sent him, but after thorough examination came to the conclusion that it was quite insufficient to account for the action produced. The quantity of copper carried into the boilers being so small could only account for a proportionate decay of the iron of the boilers, which was comparatively unimportant. He concluded with Mr. Ogston, that the corrosion of the iron of the boilers was entirely due to the action of the acids separated from the tallow by the process known as water saponification.

Ordinary tallow is a compound of fatty acids, chiefly stearic, with a peculiar basic substance called, from its sweetness, glycerine. When tallow is boiled in a solution of soda it is decomposed into stearic acid and glycerine. The stearic acid unites with the soda, forming a soap, the glycerine remaining free. A similar decomposition takes place when tallow is boiled alone at high pressures and temperatures, such as prevail in steam boilers, but in this case there is no soda to combine with the acid, which remains free; and though these fatty acids are feeble in comparison with mineral acids (such as sulphuric, hydrochloric, &c.), yet they possess corrosive properties, and are capable of slowly attacking iron and other metals.

One of the most important remedies recommended by Professor Hofman was the employment of hydrocarbon or mineral oils for the lubrication of the cylinders, slides, &c., instead of tallow or oils of vegetable or animal origin, as these mineral oils are not subject to decomposition so as to produce fatty acids. Within the last few years the use of mineral oils for the internal lubrication of engines with surface condensers has become general.

In 1873 Dr. Percy made a report to the Admiralty on the subject, in which he stated that the whole case might probably be accounted for by the fact that water which is approximately pure, like that condensed from steam, exerts a sensible corrosive action on iron. He instanced the boilers and pipes used for warming the Houses of Parliament, which had no copper or brass connections, and were worked with fresh water, without any admixture of grease of any sort, but which had suffered considerable corrosion, though they obviously could not have been affected by copper or fatty acids.

Dr. Percy showed no favour to the theory that galvanic action was set up by the influence of the copper in the condensers, &c. He considered that the practice of admitting salt water to the boilers would probably have a beneficial effect.

The opinion of Dr. Percy, as to the corrosive effect of distilled water, does not appear to be corroborated by the general experience of engineers as to the action of surface condensation in the ordinary working of marine boilers; and as a rule it is found to be desirable to fill the boilers with fresh water on starting, to change it as little as possible, and to make up the unavoidable losses with fresh water if such be available, the double bottoms of the ships being sometimes utilised for the storage of fresh water for this purpose.

The full reports of these and other eminent and experienced chemists are published with the Reports of the Admiralty Committee on Boilers, and they clearly show how difficult it is to arrive at correct conclusions even as to the chemical part of the question.

One of the most remarkable features of this corrosive action, more particularly when surface condensation was first introduced, was the irregularity of its action. Whilst the boilers of some ships were completely worn out in a

very short space of time, the boilers of other ships employed on the same service, and treated apparently in a similar manner, showed no sign of unusual corrosion. Even in the same boiler some plates have been found to be seriously corroded whilst adjoining ones have been unaffected.

The most serious manner in which the special decay that appeared to result from surface condensers showed itself was in the form of *pitting* or local corrosion. From this cause in many cases deep pits have been rapidly formed in the plates, with edges almost as clearly defined as if they had been punched or drilled. This action is most difficult to explain, but it has generally been attributed to the presence of slag, or to irregularity in the structure of the material, the softer parts being the more readily attacked.

So far as our present knowledge of the subject goes, the most probable causes affecting the decay of boilers would appear to be:

1. The quality of the materials and workmanship employed in their construction.

2. The length of time new boilers remain in store or in reserve before being actually brought into regular work, especially if the boilers have been previously worked on the full power trials.

3. The deterioration that ensues from the ordinary work on active service.

4. The special corrosion that sometimes ensues when boilers are fed with water from surface condensers.

5. The treatment the boilers receive from the officers in charge.

Unless skill and care be exercised in the manufacture of any structure its strength and durability will be decreased. Inferior workmanship may be prevented by efficient supervision.

In boiler work great care should be exercised to insure

that the holes in the plates at the joints should be *fair* before the rivet is put in. If, when the plates are put together, the rivet-holes are found to be not exactly true or fair, they should be made so, by the use of a rimer to cut away the uneven portions, and not roughly drifted into a semblance of truth. In cheap and careless boiler work it is often found that this drifting is so severe, that the plate is actually broken in manufacture, and it is obvious that in such a case durability cannot be expected. Sometimes, when the holes are not quite fair, the workmen have been known to use a smaller rivet in order to get it in and hide the fault instead of remedying it. The worst feature of these defects is, that they cannot generally be discovered when the boiler is made, and only show themselves after the boiler has been subjected to the strain of actual work, when it is very difficult if not impossible for them to be effectually rectified.

It is also important that the strength of the plate at the joint should not be unduly decreased by punching. As far as possible the plates of a boiler should always be fitted together and marked out in place for punching or drilling, so that the rivet holes, when the plates are finally put together, may be as true as possible, and require little or no work to be done to any of them with a rimer.

In the principal boiler-works the rivet-holes in the plates are drilled much more frequently now than was formerly the practice. There has been much discussion as to which of the two systems is preferable on the whole, but we think that on the ground of efficiency, especially for the thicker plates now used for high-pressure boilers, drilling is far superior to punching, and the question simply resolves itself into one of expense.

When plates are punched from the faces that have to come together at the joint, the holes taper outwards, so that the part of the rivet in the holes will be coned

roughly and tend to keep the plates together independently of the action of the heads and points. When the holes are drilled, this counter-sinking must be done by a tool, which involves additional expense. The plates, however, can be drilled together, and thus insure the rivet-holes being perfectly true and fair; and the strength of the plate is not diminished by drilling as it is by punching. When plates are drilled together they should be taken apart before being riveted, to allow the burr or sharp edge to be taken off from the holes. With suitable drilling machinery and plant, the cost of drilling very little if anything exceeds that of punching, and the results are much more satisfactory.

Another point to be secured is to get the joints properly closed, so that little caulking is necessary. In good boiler-making, the joint should be tight even without caulking; and if it be not fairly tight no amount of caulking will remedy it, though it may gloss over the defect for the time. Excessive caulking is very injurious, and by cracking the skin and indenting the plate at the joint is probably one of the most fruitful causes of the *grooving* that sometimes occurs along the rivet-seams. It also tends to raise the edge of the upper plate and cause looseness at the joint. The edges of the plates should be planed with a slight bevel, and the only caulking required should be a little along the thinner edge of the bevel.

The materials used in the construction of boilers, whether iron or steel, should be of as high a quality as possible, and too much care cannot be exercised in its manufacture. This is equally true both with respect to its strength and durability. The most difficult point to decide is as to the nature of the material that is best suited for the purpose. Till within a comparatively recent period wrought iron was universally employed, and it was supposed that the irregular manner in which the corrosion

showed itself in iron boilers might be accounted for by the manner in which the plates were made.

Puddling.—Wrought iron is produced from cast iron, by the process of puddling, which is performed in a reverberatory or air furnace. About five or six cwt. of pig iron, broken into pieces, is placed on the hearth of the furnace with about one-fifth its weight of hammer scale (black oxide of iron) and the whole melted together.. The oxygen from the air and from the hammer scale combines with the carbon in the cast iron, forming carbonic oxide, which passes off. The puddler, with an iron bar or paddle passed into the furnace through a small working door about five inches square, stirs up the molten metal, to facilitate the absorption of oxygen, and, after a time, small clotted lumps of the purified iron separate, or *come to nature*, in the molten mass.

The temperature at which the furnace is kept plays a most important part in the operations. While the carbonic oxide is passing off, the temperature is kept comparatively low, for if the metal were in too thin a condition, it could not be so well incorporated with the oxide used to purify it. When the disengagement of the carbonic oxide has nearly ceased, the temperature is raised, so as to soften the lumps of purified iron and enable them to be welded into a compact mass. The puddler, by working his iron paddle, rolls the metal about over the hearth of the furnace, until he forms it into five or six balls, each about sixty pounds in weight, from which he squeezes the slag as well as he can to make them more solid. The working door is then closed by a brick, and the furnace raised to a full welding heat; each puddle ball is then lifted out of the furnace by an iron rod or porter, which is pressed into it, and, under the steam hammer, quickly knocked into a rectangular block. It is then passed through rolls several times in succession, till it is formed

into a slab about one and a half inches thick, ten or twelve inches wide, and from two to three feet long. By this process the slag is to a greater or less extent squeezed out, and these slabs are known by the name of puddle bars. Some of these puddle bars are again heated and rolled, and are then known as twice-worked iron.

Staffordshire plates.—In order to make a plate of what is known as best best Staffordshire iron, a pile is made of six slabs, four of puddle bar, one and a half inches thick, and two of twice-worked iron, half an inch thick, the two slabs of twice-worked iron being placed under one of puddle bar. This pile is heated to a welding heat in a reverberatory furnace, the time required being from one and a half to two hours, and is then rolled out into a plate.

In some superior brands of iron for Admiralty boiler plates, designated best best best plates, the piles consist of four slabs of twice-worked iron and two of puddle bar, the puddle bar being in the centre of the pile. This tends to make the plates stronger and with smoother surfaces. The puddle bar being so much more fusible than the twice-worked iron assists the welding and tends to make the plate solid. If the piles were made without puddle bar the plates would be laminated, as the double-worked iron does not weld so readily. The twice-worked iron gives tenacity to the plate, the puddle bar its bending and welding properties.

Yorkshire plates.—The process of manufacture of Yorkshire iron plates, which were always used for the furnaces, combustion chambers, and other similar parts of boilers, on account of their great ductility and high welding and flanging properties, is more elaborate than that just described. No doubt the superiority of Yorkshire iron is to a great extent due to the high quality of the ore and of the coal employed for smelting, but something must be

allowed for the extra care and expense displayed in its manufacture. The pig iron is refined before it is puddled, and this process, whilst it adds considerably to the expense of manufacture, gets rid of many impurities.

Refining.—The process of refining consists in melting the pig iron in an open oblong trough made of cast iron, three sides of which are enclosed in double walls and kept cool by the circulation of water between them. The hearth is filled with coke, pig iron is then laid on, and coke heaped up over the whole. The tuyeres for the air blast are inclined at an angle of 20 or 30 degrees to the floor of the hearth, so that the blast may be directed on the metal. When the furnace is lighted the air blast soon raises the temperature sufficiently to melt the iron, and partially oxidises it. The silicon contained in the pig iron unites with the oxygen in the oxide, forming silica, which passes off in the slag or *finery cinder*. Four or five cwts. of coke are required for the refining of a ton of pig iron, fresh coke being added as required during the process. In about two hours the *fine metal* is drawn off into a shallow cast-iron trough, lined with loam, which is kept cool by the circulation of water below it. The bulk of the slag runs off into a separate mould. The slab of fine metal is about two inches thick, and is chilled by pouring water on it. This makes it very brittle, and enables it to be easily broken into pieces suitable for the puddling furnace. The refining process extracts nearly the whole of the silicon from the iron, and diminishes the proportions of carbon, manganese, sulphur, and phosphorus.

In working up the puddle bar for making Yorkshire plates much more extended use is made of the hammer than for ordinary boiler-plates. The whole of the work is done under the hammer until the pile is welded into a slab large enough to make the plate; this slab is then

passed through the rolls to form the finished plate. The piles for these plates are made entirely of well-hammered puddle bar, the grain being well crossed. The process of manufacture by rolling entirely, as previously described, is more expeditious, but the slag is not so effectually expelled as when the piles are repeatedly worked under the steam hammer.

It will be seen from the methods of manufacture of iron plates that homogeneity cannot be expected. The slag is squeezed out to a greater or less degree according to the amount of work performed on the iron, but it is impossible to be certain that it has been altogether eliminated.

Even with the Yorkshire plates, on which the greatest amount of work and care are exercised, the welding of the slabs may be imperfect in some parts, causing laminations. A small piece of slag or cinder getting between the slabs would spread out during the rolling and cover a considerable space, preventing welding at that place, and producing lamination, which would probably show as a blister if the plate were used in the furnace of a boiler. These defects are difficult to discover, as the plates look sound on the outside. If near the surface these laminations may be detected by the difference of sound in lightly tapping the plate with a small hammer.

Steel plates.—With steel the case is quite different, as this material can be cast in an ingot of sufficient size to form the plate, so that no welding is required, and the ingot has only to be hammered and passed through the rolls to form the finished plate, the structure of which should therefore be homogeneous.

For some years steel was looked upon with suspicion by engineers and regarded as treacherous and unreliable for boilers. The steelmakers have, however, now succeeded in producing a material which, whilst containing a suffi-

cient percentage of carbon to enable it to be fused to insure homogeneity, cannot be hardened, and may be worked with as much freedom as iron. The mild steel made by the Siemens-Martin process has now come into general use for boiler work, and has to a great extent removed an important part of the difficulty connected with the boiler corrosion question. It is stronger than iron, and therefore boilers may be made of less weight to sustain the same pressures; the plates can be made at less cost and of much larger size than the superior iron boiler-plates, which lessens the expense and simplifies the construction of boilers in many cases. Modern marine boilers are made entirely of steel, the furnaces and internal parts that have to be welded and flanged being made from plates of special soft quality.

Boilers remaining in store, or on board ships in the Reserve, for a lengthened period before being used, if proper precautions be not taken, may suffer considerably, and deteriorate very rapidly after being set to work. Formerly it was considered sufficient to paint them inside and keep them open, occasionally airing them by means of fires in the ash-pits. Even the painting inside was not in all cases deemed necessary if the boilers were kept fairly dry. It was, however, found that the moisture in the atmosphere produced oxidation of the plates and stays, which, though slow, if long continued, caused serious decay before the boiler was set to work at all.

To prevent this, boilers in store are now carefully dried, pans of quicklime placed in them, and the boilers closed up, being, in fact, hermetically sealed, to prevent access of air. The quicklime will rapidly absorb any moisture that should remain in the boiler. This plan has proved most successful, the boilers being opened out and the lime renewed about twice a year.

When boilers are in ships in the Reserve, the danger

of leakage through the cocks and valves renders another system desirable. At first they were entirely filled with a solution of lime. This was successful in preserving the boilers, but the lime deposit at the bottom was objectionable, choking the cocks and pipes, &c. The plan now frequently adopted is to fill the boilers quite full with a solution of carbonate of soda, containing, say, about 50 pounds of common soda to 100 cubic feet of water. This achieves its object, and the bottoms of the boilers are free from sediment, the soda remaining in solution. The boiler is filled by putting on a slight pressure, and allowing the air to escape through a small cock and pipe fitted for the purpose at the highest part of the boiler. If possible the water should be heated to assist the expulsion of the air.

One objection raised to this practice is that the cold surfaces tend to condense the moisture on the exterior parts of the plates, which may cause them to oxidise somewhat outside. The moisture may also settle on the skin and beams of the ship in the vicinity of the boilers and assist their decay.

If the dry lime process be adopted for ships afloat, special precautions should be taken to prevent leakage. The cocks and valves should be carefully fitted, well tallowed before replaced, and then screwed down tight. The extent of the preservation of boilers that have been closed up with dry lime may be tested when the boilers are opened, by the simple process of putting a light inside. If there has been any oxidation the air will be impure and the light will burn feebly or be extinguished, according to the amount of the decay, but if the plates have not suffered, the light will burn steadily.

Another method of preserving the boilers when not in use, which may be readily applied, is to place pans of burning charcoal, or coked coal, inside the boiler, which

should then be closed up to exclude the air. The charcoal will burn until a large percentage of the oxygen contained in the air in the boiler combines with the carbon, forming carbonic acid gas, and there will then be very little free oxygen remaining to act on the plates and stays. Pans of quicklime should be placed in the boilers as well as the burning charcoal, in order to absorb the moisture.

This plan is very suitable for the cases of ships in commission, in which it is not always easy to find time to thoroughly dry them and keep them perfectly free from moisture. By the burning of the charcoal, nearly all the oxygen is absorbed, and the small quantity remaining will be insufficient to produce much injury even if the boiler be not perfectly dry; as in the absence of free oxygen, moisture cannot produce oxidation.

In cases where the circumstances do not admit of any of the foregoing methods for the preservation of the boilers being adopted, the boilers may be run or pumped up, so as to be *quite full* of water, to prevent the oxidation of the plates and stays by the action of atmospheric air on them.

The next point to consider is the deterioration of boilers from the ordinary work on active service. This is a point specially affecting the boilers of war-ships, on account of the great variety of work performed and of its intermittent action. Frequently, after steaming for a short period the fires are banked or drawn and the ship proceeds under sail. Again, war-ships on ordinary service only steam at slow speeds, the engines developing, say, from one-fifth to one-tenth their full power. Only a portion of the boilers are required at one time, and unless great care is exercised, moisture will get into the boilers not in use, through the cocks and valves, and this moisture, if atmospheric air be present, is one of the most fruitful causes of corrosion. The main boilers are often

used for distilling purposes, the steam only being kept up for a short time, and the fires probably drawn or banked at night.

This intermittent state of action and rest is very injurious to the boilers. The consequent recurring expansion and contraction cause the scale to be detached, and leave the bare surface exposed to fresh corrosive action. The interior surfaces would often be in a damp state and exposed to the action of the atmosphere, which are the most favourable conditions for the process of oxidation. It is from the nature of the service that the lives of naval boilers have sometimes been shorter than those of ocean steamers in the mercantile marine, though the latter do much more work, but under more regular circumstances. The steady continuous steaming of boilers in the mercantile marine is much more favourable to durability than the irregular intermittent work of boilers in the Royal Navy.

It is difficult to provide remedies for this cause of decay, for the work of war-ships must necessarily be irregular; but a few general principles should be borne in mind so as to reduce the evils to a minimum. In the first place, changes of temperature should take place as seldom as possible, and then the change should be gradual; sudden changes should always be avoided. Unless in cases of great emergency steam should not be raised rapidly, but as slowly as possible, to allow the different parts of the boilers as they become heated to gradually expand and prevent local straining. When fires are once lighted in a boiler, it should be kept at work until steam is finally dispensed with, and banking fires should be resorted to as rarely as possible, to prevent change of temperature. Any little saving of coal that may be gained by frequently banking fires is dearly paid for at the expense of the boilers themselves. Very much less injury is done to

boilers by steaming steadily for some time than by raising steam frequently during the time.

Considerable injury may be done to boilers by the practice of drawing fires when steaming is over. During this process large volumes of cold air rush through the boilers, and the sudden contraction that ensues, frequently causes leaks and damages the boilers. In all cases, if possible, sufficient notice as to the probable length of time that steam will be required should be given to the engineer, who could then allow the fires to burn down; and when the engines were done with, everything on the boilers might be closed up, and all allowed to cool down together, so that the boilers might contract gradually and prevent undue strains coming on any part. The furnaces could be cleared out after all had become cold.

The water should never be blown out of a boiler by the steam pressure, unless on an emergency, but allowed to remain until it is cold, when, if necessary, it should be pumped out. This is a more tedious process, but the efficiency of the boilers is the first point to be considered.

Every precaution should be taken to keep the stop and feed valves on the boilers not in use quite tight, so that the surfaces of these boilers may not be made moist from the action of the water and steam in the working boilers.

The special corrosion due to the use of surface condensation was discussed in the earlier part of this chapter. It was attributed mainly to the action of the fatty acids evolved from the tallow and vegetable oils used for lubrication, and this has been to a great extent remedied by the use of mineral oils for internal lubrication, as these oils do not decompose and form acids. The deterioration that still occurs appears to be principally due to the action of air, carried in with the feed-water when the

boilers are at work, or combined with moisture when they are empty.

Zinc protectors.—As, however, it has not yet been absolutely proved that galvanic action in some form does not exist, slabs of zinc are suspended in various parts of the boilers, both below the water-line and in the steam space, as an additional precaution. If there be any galvanic action the zinc slabs will be attacked instead of the iron of the boiler itself. It is important in fixing these zinc slabs that they should be in actual metallic contact with the iron of the boiler; if not, they are useless. When properly fitted they produce a beneficial effect, though the destruction of the zinc does not prove that in its absence galvanic action between the iron of the boiler and any superior metal would have taken place. The zinc and the iron being in metallic contact in the water of the boiler form a battery in themselves, and the zinc forming the positive pole, dissolves. The arrangements of the zinc slabs as generally fitted in boilers are shown in Figs. 19 to 27.

The interior surfaces of new boilers are often coated with a thin layer formed of three coats of Portland cement, in order to protect them from corrosion. To insure the adhesion of the cement it must be fresh ground, and laid on with a brush like paint, care being taken that each coat is perfectly dry before the next is laid on. If the interior surfaces of the boilers could be completely coated with a substance impervious to the action of water and air they would be perfectly protected. The nearest practical approach to this appears to be a thin coating of hard scale from sea-water, which can be readily obtained by careful working of the boilers, the scale not being allowed to attain sufficient thickness to prevent the passage of heat.

In some ships appliances have been fitted to allow lime

water to be pumped into the boilers with the feed-water, to neutralise any acids that might exist in the water of the boiler. The system was introduced by M. Hétet, and was tested in the French Navy and favourably reported on. It has also been applied to a few ships in the Royal Navy in the following manner:—A small tank was fixed in the engine-room near the feed-pumps. To this tank two cocks and pipes were fitted, one connected to the delivery side and the other to the suction side of the feed-pump. The former was for the purpose of filling the tank when necessary, and the other to allow the lime water in the tank to be pumped into the boilers with the feed water. Common slaked lime was put into the tank at intervals, say every half-hour, the more continuous the better, and the quantity required was about one-quarter the weight of lubricants used for the cylinders, slides, &c.

It is probable that it would be advantageous if the feed-water were filtered before being allowed to enter the boiler, so as to free it from impurities, but it is very difficult to arrange this on the large scale that would be necessary on board ship, and it has not been attempted to any considerable extent. In the few cases, however, in which it has been tried, the results have been satisfactory, and many substances that would possibly have had a deleterious influence on the boiler have been arrested or neutralised by the action of the filter.

With reference to the treatment boilers receive from the officers in charge little need be said. It is sufficiently obvious that the durability of a boiler must depend greatly upon whether its treatment has been careful and intelligent, or otherwise. The human factor in this question cannot be ignored.

The various causes affecting the durability of boilers, so far as our present knowledge goes, have been discussed and remedies suggested. The rest must be left in the

hands of those in charge, and the Steam Manual which is now issued to all commanding and engineer officers of Her Majesty's fleets contains precise detailed instructions which, if carefully studied and intelligently carried out, must necessarily lead to beneficial results with respect to the durability and continued efficiency of the boilers.

PART III.

THE STEAM.

CHAPTER IX.

EFFICIENCY OF THE STEAM.

THE total amount of energy in the form of heat transferred to the water in the boiler in order to convert it into steam is not given out in mechanical work in the engine, but only a small portion of it, say, in ordinary cases, from one-twentieth to one-fifth, according to the type of engine. The ratio which the mechanical work done by the steam bears to the total amount of energy contained in it is called *the efficiency of the steam.*

It was pointed out in Chapter IV. that the total heat of steam increased as the temperature of evaporation was raised. In other words, the expenditure of heat necessary to produce a given weight of steam from water supplied to the boiler at a given temperature, increases when the pressure and temperature of the steam are increased. The rate of increase in the total heat of evaporation is, however, very slow. For example, the expenditure of heat required to produce a given weight of steam at the pressure of 10 atmospheres is only 1·04 times that necessary to produce an equal weight of steam at the atmospheric pressure, the temperature of the feed-water in each case being 100° Fahr.

Since the difference between the amounts of heat required to produce a given weight of steam at different pressures is so small, the problem of obtaining the greatest possible quantity of work from a given expenditure of heat is reduced practically to the simpler one of obtaining the greatest amount of work from a given weight of steam, the difference in the total heat of evaporation at various pressures being so slight that it may be neglected in practice.

Mechanical work is produced by the exertion of a force through a space, and the mean value of the force, multiplied by the space through which it acts, gives the amount of mechanical work done. In the case of a steam-engine the space is represented by the distance through which the piston travels in a given time, and the force is the excess of the forward pressure exerted by the steam on one side of the piston, during its admission and expansion, above the back pressure it exerts on the other side of the piston when it is passing into the atmosphere or condenser, according as the engine is non-condensing or condensing. In a non-condensing engine the back pressure is generally from 3 to 6 pounds per square inch above the atmospheric pressure, or, say, from 18 to 21 pounds per square inch absolute. In cases of quick-moving engines with cramped exhaust passages it is sometimes as high as 10 to 12 pounds per square inch above the atmosphere. In a well designed condensing engine the back pressure is from 3 to 5 pounds per square inch absolute, that is from 12 to 10 pounds per square inch below the atmospheric pressure. The difference between the back and atmospheric pressures is usually shown in inches of mercury and is technically called the *vacuum*.

In an expansive engine, the forward pressure exerted by the steam has to be divided into two parts, viz.: the pressure during admission, whilst the steam is passing

direct from the steam pipes into the cylinder; and the pressure during the expansion of the steam in the cylinder after its admission has been cut off. The action of the steam in the cylinder may be best explained by means of a diagram, in which the horizontal ordinates represent volumes and the vertical ordinates pressures. For a cylinder of given diameter the volume may be represented by the length of the stroke of the piston. This diagram we will call the theoretical indicator diagram. Its area may be computed from geometrical principles, so that it may be used in theoretical investigations on the power and efficiency of the engines.

In Fig. 56 let O P represent the stroke of the piston of the engine, and O A the absolute pressure of the steam

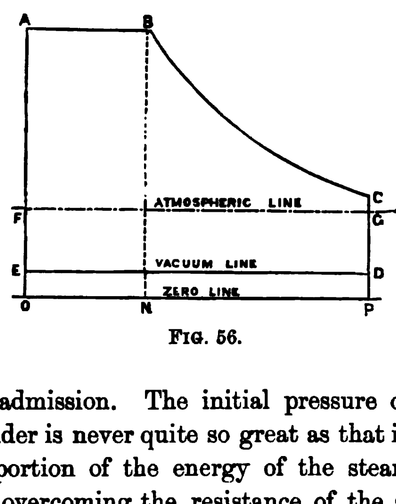


FIG. 56.

during its admission. The initial pressure of the steam in the cylinder is never quite so great as that in the boiler, because a portion of the energy of the steam has to be exerted in overcoming the resistance of the steam pipes, ports, passages, &c. In ordinary cases the reduction of pressure due to this cause may be taken to be about one-twelfth of the absolute pressure of the steam in the boilers.

During admission the steam passes into the cylinder at this slightly reduced initial pressure, and this part of

the action of the steam is shown by the line A B, which is commonly known as the steam line of the diagram. At B, or, in other words, when the piston has traversed the part O N of its stroke, the whole stroke being represented by O P, the admission of steam to the cylinder is cut off by the closing of the steam ports by the valve.

The expansion of the steam in the cylinder now commences, and the piston is pressed forward by the expansive force of the steam, the pressure continually diminishing as the piston moves onward to the end of the stroke and increases the volume occupied by the steam. This part of the action is represented by the expansion curve B C, $\frac{\text{O P}}{\text{O N}}$ being the ratio of expansion.

The laws according to which the pressures of steam diminish during its expansion vary under different conditions of heat and initial states of the steam. For practical purposes, however, it is sufficiently accurate to assume the simple approximate law that the pressure will vary inversely as the volume; that is, when the volume is doubled the pressure falls to one-half, when it is trebled to one-third, and so on.

When the piston arrives at the end of its stroke and the expansion is finished, the communication with the condenser is opened, the steam rushes off to the condenser, and the pressure falls to P D, the constant condenser pressure, which back pressure acts against the piston during the whole of the return stroke. The line D E is technically called the vacuum line of the diagram and represents the back pressure on the piston during the stroke.

O P is the zero line, or line of no pressure, from which all absolute pressures are measured. F G is called the atmospheric line, O F representing the pressure of the atmosphere. It is important to remember that in all investigations on the action of steam, the *absolute pres-*

sures, or the pressures measured from the zero line, must be taken, and not the pressures indicated by ordinary pressure gauges, which are only comparative, and simply represent the excess of the steam pressure above the pressure of the atmosphere.

It is clear from the diagram that work would continue to be done by the steam whilst its forward pressure was greater than the constant back pressure P D; so that in order to get the greatest possible quantity of work out of the steam, the expansion should be continued until the forward pressure became so far reduced as to be just equal to the back pressure; that is, the expansion curve should touch the back pressure line D E, so that there should be no sudden fall of pressure on admission to the condenser.

In a steam-engine a portion of the work of the steam is expended in overcoming the friction of the working parts of the machinery, so that, theoretically, in order to obtain the greatest possible amount of *useful* work from a given quantity of steam, the expansion should be carried out until the pressure in the cylinder was so reduced as to be just equal to the back pressure *plus* a pressure equivalent to the friction of the mechanism.

In practice it is not possible to carry out the expansion, efficiently, to so great an extent as this, and it must only be taken as a theoretical statement of what might be the case if the steam were expanded in a perfectly non-conducting cylinder, and as the condition to which we must endeavour to approximate as closely as possible by suitable appliances to existing engines.

It is obvious that as the proportion in which the forward pressure exceeds the back pressure is increased, the greater will be the amount of work performed by a given weight of steam. To take an extreme case, on the principle of *reductio ad absurdum*, if the forward pressure be just equal to the back pressure no work at all is performed,

and if it be only equal to the back pressure *plus* the pressure equivalent to the friction, no useful work is done, as all the power is expended in overcoming the friction of the machinery. Hence the higher the forward and the lower the back pressure the greater is the efficiency of the steam. Since the pressure of saturated steam varies with its temperature, this principle may, in the case of a condensing steam-engine, be expressed in the following manner, viz., the temperature of the steam on its admission should be as high as possible, and the temperature in the condenser, on which the back pressure depends, should be as low as possible.

This principle was first distinctly stated by James Watt, and he invented the separate condenser as a means of carrying it into effect. The only limit to the initial temperature and pressure of the steam is the consideration of the strength and safety of the boiler and cylinder. In Watt's time, workmanship and knowledge of the strength of materials were not in such an advanced state as at present, so that most engineers of that time were necessarily very cautious in the adoption of high pressures, and relied more on obtaining a low back pressure. As experience was gained, the pressures at which boilers were worked were gradually increased, and of late years the advances in that direction have been great, and there has been much gain in economy from the high pressures and rates of expansion now in general use.

The following table shows how the amount of work done by one pound of steam is augmented as the initial pressure and rate of expansion are increased. Since the total heat of steam is practically the same at all temperatures, the increase in the performance of work may be taken to represent very nearly the theoretical increase in efficiency due to the increased expansion. The steam is supposed to remain in a state of saturation during its ex-

pansion, and its pressure in each case at the end of the expansion is supposed to be the same, viz. 10 pounds per square inch absolute. The back pressure in each case is taken at 3 pounds per square inch absolute.

Initial absolute pressure	Relative volume	Specific volume	Ratio of expansion	Mean absolute pressure	Mean effective pressure	Relative indicated horse-power
10	2368	37·8	1·0	10·0	7·0	100
20	1231	19·7	1·9	17·2	14·2	289
40	643	10·3	3·6	24·8	21·8	549
60	434	7·0	5·2	29·4	26·4	732
80	338	5·4	6·8	33·6	30·6	914
100	270	4·3	8·4	36·0	33·0	1024
200	142	2·27	15·5	44·0	41·0	1417

The mean absolute pressure is the average of the absolute forward pressures during the stroke, and the mean effective pressure is equal to the mean absolute pressure minus the assumed constant back pressure of 3 pounds per square inch.

Relative volume.—By relative volume is meant the ratio of the volume of the steam produced to that of the water from which it was generated.

Specific volume.—The specific volume of steam is the volume, in cubic feet, of one pound of steam at any given pressure.

It will be seen that by increasing the initial pressure of the steam from 20 to 80 pounds per square inch absolute the work done per pound of steam is increased more than three times, whilst at 100 pounds pressure it is three and a half times, and at 200 pounds nearly five times as great as at 20 pounds pressure absolute, or, say, 5 pounds above the atmospheric pressure, which was the ordinary working pressure in the early days of steam navigation, the heat required to produce the steam being practically the same in each case.

This great increase in the efficiency of steam by utilising its expansive powers may be shown graphically

by means of ideal indicator diagrams in the following manner. In Fig. 57, O P represents the zero line, or line of no pressure, the length O P being taken to represent the volume of the steam at the end of the stroke. Pressures are measured by the vertical ordinates, the scale of the diagram being 40 pounds to the inch, *i.e.* an ordinate one inch long represents a pressure of 40 pounds per square inch and so on. We have taken for illustration the cases

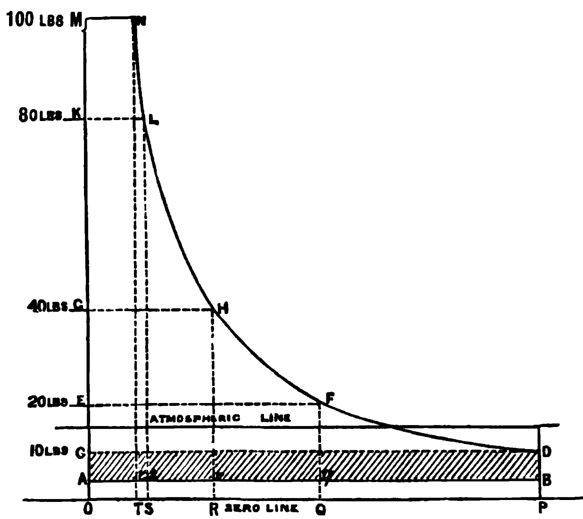


FIG. 57.

in which the initial absolute pressures are respectively 10, 20, 40, 80, and 100 pounds per square inch.

O A represents a pressure of 3 pounds, which is the assumed constant back pressure, and A B, drawn parallel to O P, represents the back pressure or vacuum line of the diagrams.

O C represents a pressure of 10 pounds absolute; and the shaded diagram C D B A represents the work done by the steam without any expansion.

O E represents a pressure of 20 pounds absolute, or,

say, 5 pounds above the atmospheric pressure. This may be taken to represent the pressures of steam used in the earlier marine boilers and engines. In this case the ratio of expansion is 1·9, and the admission of steam is cut off at the point F, the stroke being completed by the expansion of the steam. The indicator diagram representing the work in this case is A E F D B A, the part done by direct pressure during admission being shown by the rectangle E F q A, and that due to expansion by the area F D B q.

O G represents a pressure of 40 pounds absolute, or, say, 25 pounds above the atmosphere, which may be taken as the ordinary pressure used in marine engines prior to the introduction of compound engines and high-pressure steam. The rate of expansion in this case is 3·6 times, the theoretical indicator diagram being A G H D B A.

O K represents a pressure of 80 pounds absolute. This is the case with the earlier compound marine engines, the pressure being about 65 pounds per square inch above the atmosphere. The work done is shown by the diagram A K L D B A, and it is easily seen that this is a great improvement even on the indicator diagram with steam of initial pressure of 25 pounds above the atmosphere. The steam in this case is expanded 6·8 times; the work done by direct pressure being shown by the rectangle K L S A, all the rest of the work being due to the expansion of the steam.

The diagram A M N D B A represents the work done with steam of an absolute pressure of 100 pounds per square inch, or about 85 pounds above the atmosphere, the assumed rate of expansion in this case being 8·4.

The several diagrams just discussed show very clearly the gain in economy that has resulted from the successive steps taken from time to time in increasing the initial pressure and ratio of expansion of the steam, the quantity of heat expended in evaporation being approximately the same in all the cases, though the amount of work done at

the highest pressure shown is more than ten times as great as when no expansion was employed and all the work done by direct pressure.

The economy due to expansive working of the steam may also be practically illustrated by taking indicator-diagrams from the engines when working expansively and throttled respectively. First take a diagram with the expansion gear in use, carefully noting the exact number of revolutions per minute that the engines are making at the time. Then throw the expansion gear out of operation, and partially close the throttle valve, until the revolutions

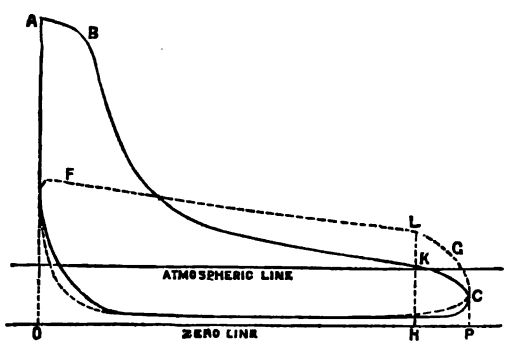


FIG. 58.

are the same as before. If the two diagrams be taken on, or transferred to, the same card it will be found that they will be as shown in Fig. 58. The revolutions being the same in the two cases, the work done will be the same, and consequently the areas of the two diagrams must be equal to each other. The expansion diagram is represented by the full lines and the throttled diagram by the dotted lines. The pressure line, F G, of the throttled diagram will therefore be below that of the expansion diagram, A B C, at the commencement, and above it at the end of the stroke. The quantity of steam used is represented by its pressure at the end of the stroke, when the

cylinder may be considered to be full of steam at its final pressure, and we therefore see that the amount of steam required to perform a given quantity of work when used expansively is less than when it was throttled and no expansion employed. O P represents the full stroke of the piston. If we take any point, H, of the stroke just before release commences, it will be seen that the quantities of steam required per stroke, when working expansively and throttled, will be proportional to the absolute pressures, H K and H L, respectively. As less steam is used per stroke when working expansively than when throttled, the vacuum in the former case will be better, unless the quantity of condensing water is increased, which would augment the work done by the pumps, and thus further decrease the efficiency of the engine.

The following table, abstracted from the results of a series of experiments made on the engines of the United States ships 'Bache' and 'Dexter,' shows in a quantitative form the gain in economy that follows the use of high-pressure steam worked expansively as compared with steam at a lower pressure worked at a reduced rate of expansion. The trials were very carefully made and complete in their character, and further analyses of the results obtained will be given further on. The diameter of the cylinder of the 'Bache' was twenty-five inches, with a stroke of two feet. Diameter of 'Dexter's' cylinder, twenty-six inches, with a three feet stroke.

	'Bache'			'Dexter'	
	1	2	3	4	5
Pressure in boilers lbs.	81·07	80·83	30·87	68·7	39·9
Ratio of expansion	8·57	12·62	2·18	4·46	2·42
Vacuum inches	25·29	24·66	24·0	25·86	26·0
Revolutions per minute	46·25	39·92	45·28	56·5	55·25
Indicated horse-power	74·6	54·8	66·7	186·0	162·0
Feed-water used per indicated horse-power per hour lbs.	24·09	27·11	34·03	23·86	28·94

In the case of the 'Bache' the most efficient rate of expansion with the high-pressure steam was about five times. Above this the consumption of feed-water per indicated horse-power increased. It was therefore a matter of some importance to ascertain whether, in case of reduced power being required, the initial pressure should be reduced and the steam worked at a less rate of expansion, or whether the original pressure should be maintained with a high grade of expansion. The results clearly show that the latter course is the correct one. Columns 1 and 2 give the results of working with high pressures and expansion. In number one the expenditure of feed-water is 24 pounds per indicated horse-power per hour, and in number two 27 pounds, whilst in number three, in which the initial pressure is reduced to 30 pounds per square inch, the consumption of feed-water rose to 34 pounds per indicated horse-power per hour, the indicated horse-power in this case being between those given in columns 1 and 2.

The results from the 'Dexter' were very similar. With the high-pressure steam and corresponding rate of expansion the consumption of feed-water was only 24 pounds per indicated horse-power, whilst with the reduced pressure and expansion the consumption rose to about 29 pounds per indicated horse-power per hour, the horse-power in the two cases being nearly the same.

These results are important particularly with regard to the machinery of war-ships, which when on ordinary service is generally worked at reduced power. It is clear from the figures given in the table that in such cases it is desirable, for the sake of economy, to use the expansion gear to the greatest extent of which it is capable. It must be understood, however, that the steam should not be throttled before admission to the cylinder, so as to cause its initial pressure in the cylinder to be much below

the pressure in the boiler. This is sometimes done when engines are worked at considerably reduced powers, but the practice is a wrong one, and instead of effecting a saving of fuel it increases the expenditure and unnecessarily strains the boilers and steam-pipes.

No advantage can be gained by keeping the steam pressure high and considerably throttling it before it enters the cylinders. When working at reduced powers the expansion gear should be set to the highest grade at which it can be worked, and then the boiler pressure should be kept a few pounds above the necessary initial pressure in the cylinders, with the throttle or regulating valve wide open, or nearly so. This statement, however, requires the qualification that the boiler pressure should never be kept lower than that necessary for handling the engines readily, without the aid of the vacuum in the condensers, so as to be prepared for the emergency of stopping and starting the engines.

Though it is only in comparatively recent years that much attention has been devoted to the development of high rates of expansion of steam in order to attain economical working, we find that more than a century ago James Watt indicated the gain that would ensue from the utilisation of the expansive power of steam. During the years 1764 to 1769, Watt made many experiments to determine, so far as he could, the relations that existed between the pressure, volume, and temperature of steam, and reasoning from the data he obtained he framed and published in 1769 a body of principles expressing the conditions necessary for the efficient and economical working of the steam-engine. It is remarkable to note how sound these conclusions were, and that since that time no alteration has been made in the principles, the progress having been entirely due to improvements in the details of the machinery, and to the more complete development of the

principles that has been rendered possible by improved workmanship and appliances.

It will be interesting to state Watt's principles in his own words. He says:—

'My method of lessening the consumption of steam and consequently of fuel in fire engines consists of the following principles:—

'First.—That vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in common fire engines, and which I call the steam vessel, must, during the whole time the engine is at work, be kept as hot as the steam that enters it, first by enclosing it in a case of wood or other material that conducts heat slowly; secondly, by surrounding it with steam or other heated bodies; and thirdly, by suffering neither water nor any other substance colder than the steam to enter or touch it during that time.

'Secondly.—In engines that are to be worked wholly or partially by the condensation of steam, the steam is to be condensed in vessels distinct from the steam vessels or cylinders, though occasionally communicating with them. These vessels I call condensers, and whilst the engine is working, these condensers ought to be kept as cold as the air in the neighbourhood of the engines, by the application of water or other cold bodies.

'Thirdly.—Whatever air or other elastic vapour is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the steam vessels or the condensers by means of pumps wrought by the engines themselves or otherwise.

'Fourthly.—I intend in many cases to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the same manner as the pressure of the atmosphere is now employed in ordinary fire engines. In cases where cold water cannot be had in

plenty, the engines may be wrought by this force of steam only by discharging the steam into the air after it has done its office.

'Lastly.—Instead of using water to render the pistons and other parts of the engine air and steam tight, I employ oils, wax, resinous bodies, fat of animals, quicksilver, and other metals in their fluid state.'

During the same year (1769) Watt invented the cutting-off of the admission of steam, so as to make it work expansively, but he did not use it till 1776, and only published it in 1782, when he patented it together with his invention of the double-acting engine.

Before proceeding further it will be necessary to define a few terms that will often recur, as it is important that their meanings should be clearly understood.

Saturated steam.—In all gases the density, pressure, and temperature are connected together by certain fixed laws, so that if any two of them be known, the third can be determined. In the case of steam, or any other vapour, in contact with the liquid from which it is generated, there is, for each temperature, a corresponding density, which is the greatest density the vapour can have without its being partially, or wholly, condensed into the liquid form. Consequently for each temperature there is a maximum pressure which the vapour can exert.

A vapour which is at the maximum density and pressure corresponding to its temperature is called *saturated vapour.* It is then just at the point of condensation, and any increase of pressure or decrease of temperature will cause some of the vapour to be condensed. Steam, therefore, at any given pressure is said to be *saturated* when it is at its maximum density, consistent with its remaining as vapour. Saturated steam is often called *dry steam,* because it is pure steam without any admixture of liquid water.

The relations between the pressure and temperature of saturated vapour are given by the following formula, the constants in which are deduced from Regnault's experiments. Suppose T to be the temperature of the boiling point on Fahrenheit's scale, $\mathcal{T}$ the absolute temperature of the boiling point, $= T + 461$, and

p = absolute pressure of the steam in pounds per square inch,

Then,
$$\log p = A - \frac{B}{\mathcal{T}} - \frac{C}{\mathcal{T}^2}$$

A, B, and C being constants.

The following is the inverse formula for calculating the absolute temperature of the boiling point from the pressure.

$$\mathcal{T} = 1 \div \left\{ \sqrt{\left(\frac{A - \log p}{C} + \frac{B^2}{4C^2}\right)} - \frac{B}{2C} \right\}$$

The values of the constants in the formula are, for steam,—

A	Log B	Log C	$\frac{B}{2C}$	$\frac{B^2}{4C^2}$
6·1007	3·43642	5·59873	0·003441	0·00001184

This formula is very complex in character, and its calculation would be tedious.

For practical purposes the following approximate formula may be used:

$$p = \left(\frac{T + 40}{147}\right)^5$$

T = temperature of boiling point in degrees Fahrenheit,
p = absolute pressure in pounds per square inch.

This is nearly correct for absolute pressures between 6 and 60 pounds per square inch. It may also be used for

pressures between 60 and 120 pounds, but unity should then be added to the result.

The density of a vapour is measured by the space occupied by a given weight, and the volume of one pound of saturated steam may be calculated by the approximate formula,

$$v = 0{\cdot}41 + \frac{389}{p + 0{\cdot}35}$$

where v = volume of one pound in cubic feet, or *specific volume*,

and p = absolute pressure in pounds per square inch.

This formula can, however, only be relied on for pressures of between 20 and 60 pounds per square inch. Below 20 pounds its results are too large, and above 60 they are too small.

If tables of logarithms, or of square roots, are available, the volume of one pound of saturated steam, at any given absolute pressure, may be calculated from the formula,

$$p\,v^{\frac{17}{16}} = 475$$

where v = volume in cubic feet

and p = absolute pressure in pounds per square inch.

Superheated steam.—If the steam be removed from contact with the water from which it is generated, and additional heat be applied, the pressure being kept constant, its volume and temperature increase, as pointed out in Chapter IV., and the steam becomes *superheated*; that is, it contains more heat than that necessary to keep it in a state of saturation at the given pressure. The properties of superheated steam approach those of a perfect gas.

Moist or wet steam.—If heat be abstracted from saturated steam, the pressure being kept constant, a portion of the steam liquefies and the steam becomes *supersaturated* or *moist steam*.

Expansion.—We will now consider more particularly the subject of expansion, the laws to which air and steam conform during expansion, and the case of steam expanding in the cylinders of a steam-engine. If the vessel or chamber in which any gas is confined be enlarged or contracted, the gas will still completely fill the vessel, but at an altered pressure. During the process of expansion of a perfect gas, of which atmospheric air may be taken as a type, the pressures and volumes are connected by the law that their product is always proportional to the absolute temperature, or if

$p =$ the pressure,
$v =$ the volume of one pound of the gas,
and $T =$ its absolute temperature,
then $p\,v = c\,T$, where c is a constant quantity.

If the temperature remain constant, the alteration of pressure will be in inverse ratio to the alteration of volume. For example, if two cubic feet of air at 10 pounds pressure were compressed into a volume of one cubic foot, its pressure would be increased to 20 pounds per square inch. If it were allowed to expand into a volume of four cubic feet, its pressure would be reduced to 5 pounds per square inch, and so on. This law is generally expressed by saying that the pressure varies inversely as the volume, or that the product of the pressure and volume of a perfect gas is constant.

Therefore, in this case,

$$p \times v = \text{constant}.$$

This may be shown graphically by means of the ordinates of a rectangular hyperbola referred to its asymptotes as axes, this curve representing the law of expansion of air and other perfect gases; the horizontal distances or abscissæ representing the volumes, and the vertical distances, or ordinates, the pressures. In Fig. 59, O X and O Y are two

axes drawn at right angles to each other, O being the origin of co-ordinates. Let P M represent the pressure of the gas when its volume is represented by O M, and Q N the pressure corresponding to the volume O N. Then, by hypothesis, O M × M P = O N × N Q = constant, or $p\ v =$ constant. The curve passing through a series of such points will therefore be represented by an equation of the form $x\ y = c$, which is that of a rectangular hyperbola, to which the asymptotes are axes, the ordinates representing pressures and the abscissæ volumes. The pressure corresponding to any other volume, O S, is found by drawing

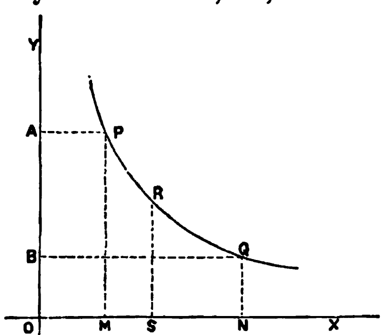


FIG. 59.

the vertical ordinate through S, cutting the curve in R, the line R S representing the required pressure. The work done by the gas in expanding from the volume O M to the volume O N is represented by the area of the figure P Q N M.

Expansion of steam.—The laws followed by steam during expansion are somewhat different from those of a perfect gas, though the general character of the expansion curve is similar.

Free expansion.—There is an important difference to be noted between the free expansion of steam, that is, its expansion without the performance of any mechanical work, and its ordinary expansion in the cylinders of a steam-engine, in which the steam during expansion exerts pres-

sure on the piston and performs work. It is necessary that this difference should be borne in mind in considering the expansive action of steam in an engine.

Imagine one pound of saturated steam at any given pressure to be confined in a cylinder behind a piston, both the cylinder and piston being conceived to be made of non-conducting materials. Suppose the piston to be moved by an *external* force, so that the volume of the steam is increased without its having performed any work on the piston. Then it is evident that as no heat has been either added to or abstracted from the steam during the process, the total amount of heat in it is the same at the end as it was at the beginning of the expansion. But, as was pointed out in Chapter IV., the total heat of saturated steam increases slowly with its pressure, so that, since the steam was saturated at its original pressure, the total amount of heat in the steam is more than sufficient to keep it in a state of saturation at the reduced pressure at the end of the expansion, so that the steam will be to some extent superheated.

To fix our ideas on this point, we will take a numerical example. Suppose one pound of saturated steam at a pressure of two atmospheres to be allowed to expand, without doing any mechanical work, to a pressure of one atmosphere. The total heat in one pound of saturated steam at the pressure of two atmospheres is 1157 thermal units. Since the steam expands without doing any external work, the whole of this heat is retained in the steam when its pressure has been reduced by expansion to one atmosphere. But the total heat necessary to keep a pound of steam at a pressure of one atmosphere in a state of saturation is only 1146 thermal units, and the difference, or 11 thermal units, must have been expended in superheating the steam.

The phenomenon of free expansion, or expansion with-

out the performance of mechanical work, is one that often occurs in compound engines on the admission of steam to the receiver or reservoir between the high- and low-pressure cylinders. The final pressure in the high-pressure cylinder is generally somewhat higher, and in many cases considerably higher than the initial pressure in the low-pressure cylinder, so that when the steam escapes from the high-pressure cylinder to the intermediate receiver, its volume is suddenly increased without any external work being done; and the difference between the amount of heat in the steam at the end of the stroke in the high-pressure cylinder and that necessary to keep it in a state of saturation at the reduced pressure is expended in drying and superheating it. This probably accounts, to some extent at least, for the fact that some compound engines which have been made without steam-jackets on the low-pressure cylinders have worked without any very great loss of efficiency as compared with similar engines fitted with steam-jackets. In these cases there has been very little expansion in either cylinder, nearly the whole of the expansion taking place on admission to the reservoir.

Work done during expansion.—When, however, steam during its expansion performs mechanical work, the conditions of the case are very different from those just discussed. We will in the first place assume that the expansion takes place in a perfectly non-conducting cylinder, so that heat is neither added to nor abstracted from the steam during the operation. Until about the year 1849 or 1850, when the true nature of heat was first determined, it had been supposed that when steam was expanded in this way the total amount of heat in it was the same at the end as it was at the beginning of the expansion. It was, however, always found that liquid water collected in the cylinders, but this was supposed to be due to priming, or the carrying of spray from the boilers to the

cylinders, which explanation was often found to be unsatisfactory.

The real cause of the presence of this water in the cylinders was, however, readily explained when the principles of thermo-dynamics became understood. It was then seen that the mechanical work done by the steam during the expansion was due to the fact that a portion of the energy that had been stored in the steam in the form of heat had become transformed into mechanical work, and appeared no longer in the form of heat, so that the total quantity of heat contained in the steam had been diminished. The abstraction of the amount of heat thus changed into mechanical work was sufficient not only to lower the temperature of the steam to that corresponding to its reduced pressure, but also to cause some of it to liquefy.

To take a numerical example :—

Suppose one pound of saturated steam, at an absolute pressure of 60 pounds per square inch, to expand in a non-conducting cylinder, without gain or loss of heat, pressing a piston before it, till its pressure fell to 3½ pounds per square inch absolute,

Then, in round numbers,

	ft. lbs.
Total heat in one pound of steam at 60 pounds pressure (absolute)	904,000
Work performed during expansion	157,000
Difference—or heat remaining in the steam at end of the expansion	747,000
Heat necessary to keep one pound of steam saturated at a pressure of 3½ lbs. (absolute) . . .	870,000
Deficiency	123,000

so that we see a considerable quantity of steam must become liquefied during the expansion.

Liquefaction in cylinders.—In this case, in which the steam has been supposed to expand in a non-conducting cylinder, the water of liquefaction would simply be carried to the condenser at the end of each stroke and no *waste* of

heat would ensue. Unfortunately, however, we have to deal with very different conditions, as the cylinders and pistons are necessarily made of conducting materials, and the loss from liquefaction in the cylinders will be found to result in an indirect and rather complex manner. The hot steam from the boiler enters the cylinder after it has been open to the condenser for a whole stroke, and when its temperature may be supposed to approximate to that of the steam passing to the condenser, say from 140° to 150° Fahr. It is therefore evident that a quantity of the heat in the entering steam will be expended in raising the temperature of the cylinder, cylinder cover, piston, &c., and the initial temperature and pressure of the steam will be thereby lowered. As the steam expands, a portion of it, due to the work done, liquefies, and probably exists in the form of spray, or collects on the surfaces of the cylinders, &c. This moisture considerably increases the power of the steam to absorb radiant heat. Consequently when its temperature has fallen below the temperature of the metal of the cylinder, it abstracts heat from the surfaces, partly perhaps during the latter part of the expansion, but much more rapidly during the period of exhaust, when its pressure has fallen to, say, 2 to 3 pounds absolute, under which pressure water boils at about 140° to 150° Fahr. The heat abstracted from the metal of the cylinder, piston, &c., by this agency goes direct to the condenser, and not only does no useful work, but increases the back pressure on the piston to an extent sensibly felt in many unjacketed engines. The cylinder being now in this comparatively cool state, fresh steam enters, and the temperature of the cylinder is again raised by the condensation of the entering steam, and the amount of water so formed adds to the quantity of water to be boiled off during the period of exhaust.

These processes go on at every stroke, and the quan-

tity of water in the cylinders progressively augments till it reaches the maximum quantity that can be re-evaporated during the period of exhaust. This quantity mainly depends on the difference between the temperatures of the steam at the beginning and end of the stroke, and therefore tends to become a constant quantity whatever the rate of expansion may be. The great loss that often arises from liquefaction in the cylinders of a steam-engine is therefore due to the fact that the liquid water in the cylinder, existing probably in the form of spray, acts as an equaliser of temperature, lowering the initial and raising the final temperatures and pressures, and thus decreasing the efficiency of the steam. The effect is the same as if, during each stroke, a certain portion of the steam passed direct from the boiler to the condenser, without performing any work whatever.

In some experiments on this point made by Mr. Isherwood, of the United States Navy, which were analysed by the late Professor Rankine, it was found that with an expansion of four times, the amount of steam thus wasted was actually more than that performing work, so that the expenditure of heat was more than doubled. The experiments were made on engines with unjacketed cylinders, having a piston speed of about 224 feet per minute. Common saturated or perhaps rather moist steam was used on the trials.

The results are shown in the following table:

Point at which the steam was cut off	0·93	0·64	0·40	0·354	0·25
Ratio of expansion . . .	1·07	1·50	2·50	2·82	4·0
Pounds of water used per stroke by actual measurement	5·2	4·4	3·4	3·63	3·02
Pounds of water used per stroke shown by indicator diagrams	4·42	3·15	1·84	1·62	1·17
Pounds of steam wasted per stroke by condensation, clearance, and leakage	0·78	1·25	1·56	2·01	1·85
Percentage of steam wasted .	15	28	46	55	61

The consumption in this case is enormously greater than that theoretically necessary, and shows how great the loss from liquefaction may become when no provision is made to prevent it. The case taken by Mr. Isherwood was an unfavourable one, because the speed of the piston was small and the steam, in all probability, super-saturated before it entered the cylinders at all.

We will now consider some more recent experiments made with modern engines worked with high-pressure steam, which show that even under these conditions, when the cylinder is unjacketed, the loss from liquefaction is very considerable.

The experiments were made by Mr. Emery, of New York, on the engines of the United States survey vessel 'Bache.' This vessel was fitted with a compound engine, having the small cylinder arranged above the large cylinder, the two pistons having a common rod. The diameter of the high-pressure cylinder was sixteen inches, and of the low-pressure cylinder twenty-five inches, the length of the stroke in each case being two feet. Direct steam communication was fitted to the large cylinder so that it could be worked independently, using steam of the same pressure and with the same degree of expansion as when both cylinders were working together as a compound engine.

We will confine our attention at present to the experiments that were made without the jacket in use in order to show the percentage of heat wasted. The cylinders were well covered with felt and lagging to prevent loss from radiation as far as possible. The particulars are given in the following table, and show that in engines in which no provision is made for the addition of heat to the steam during expansion, the loss from liquefaction in the cylinders is very considerable, even when the engines are worked under favourable conditions.

		Simple engine			Compound engine	
		1	2	3	4	5	6
Duration of trial	hrs.	2·05	1·983	1·80	2·133	2·066	1·833
Pressure of steam	lbs.	78·11	79·625	81·0	80·31	80·28	82·0
Ratio of expansion		5·32	7·62	11·82	5·634	6·658	9·146
Indicated horse-power		89·14	71·75	47·24	85·8	77·06	55·93
Water used per indicated horse-power per hour by measurement	lbs.	26·247	29·616	35·075	23·21	23·036	23·765
Water used per indicated horse-power per hour from indicator diagrams	lbs.	17·352	17·755	21·028	12·346	12·312	12·699
Difference or water wasted by liquefaction, &c.	lbs.	8·895	11·861	14·047	10·864	10·724	11·066
Percentage of loss from liquefaction, &c.	lbs.	33·89	40·05	40·05	46·81	46·55	46·56

The highest theoretical efficiency in the expansive working of the steam can only be realised if the cylinders and pistons are made of perfectly non-conducting materials. It is not sufficient to cover the exterior of the cylinders with non-conducting materials, which only prevent the passage of heat from the steam to the atmosphere and not the complex action which goes on in the cylinder by the abstraction of heat from the steam during the admission, which heat is again given out to the steam during the exhaust. In this process it is only necessary that the metal of the cylinder should be cooled for a very small distance below the surface. It would, therefore, be a great point gained if some non-conducting material were dis-

covered suitable for application to the interior surfaces which would prevent this alternate heating and cooling.

In 'Engineering,' of February 12th, 1875, Mr. Emery records an interesting and instructive experiment which he made on this subject. Two cylinders were constructed of precisely the same dimensions, one of glass, the other of iron. Either of the cylinders could be attached to a valve which admitted steam from a boiler to the cylinder and permitted it to exhaust into a condensing coil lying in a tub of water. The capacities of the two cylinders were proved to be exactly the same by transferring water from one to the other. When put in turn in the condition of a steam-engine cylinder, the average of the experiments showed that the iron cylinder used fully twice as much steam as the glass cylinder, which was proved by the fact that twice the quantity of water came through the condensing coil for the same number of movements of the valve. Steam of the same pressure was used in all the trials, and the experiments were many times repeated with substantially the same results.

This great difference may probably be to some extent due to the fact that the cylinders used in the experiment were so small. Small engines with unjacketed cylinders are less economical per unit of power developed than large ones. This is easily seen when we consider that the smaller the diameter of cylinder, the greater is the ratio of the surface of the cylinder, &c., which is alternately heated and cooled, to the volume of steam contained in the cylinder, so that the quantity of heat taken up by the metal of the cylinder during admission and given out during exhaust will be proportionately greater.

To take a simple case for illustration. In a cylinder one foot in diameter and one foot long the area of the cylinder surface is 3·1416 square feet, and the area of the piston and cylinder cover 1·5708 square feet, making a

total surface of 4·7124 square feet. The volume of the cylinder is ·7854 cubic foot, so that the ratio of surface to volume is 4·7124 to ·7854, or 6 to 1. In other words, there are six square feet of heating and cooling surface to one cubic foot of steam used. Now suppose the diameter of the cylinder to be doubled, the stroke remaining the same. In this case the volume is increased in the ratio of 4 to 1, the cylinder containing 3·1416 cubic feet of steam. The areas of the piston and cylinder cover are also increased in the ratio of 4 to 1, but the internal surface of the cylinder is only increased in the ratio of 2 to 1, and the total surface is 12·5664 square feet. In this case there are 12·5664 square feet of heating and cooling surface for 3·1416 cubic feet of steam, or only 4 square feet of surface for each cubic foot of steam instead of 6 square feet, as in the previous example. We conclude, therefore, that with unjacketed cylinders the percentage of loss from liquefaction will be less in large engines than in small ones, and consequently that large engines are more economical than small ones per unit of power developed, when the cylinders are unjacketed.

The law which steam follows when expanded in a non-conducting cylinder is of the hyperbolic family and is represented approximately by the equation

$$p\,v^{\frac{10}{9}} = \text{constant.}$$

The curve representing the relative pressures and volumes of steam when expanding in a non-conducting cylinder without gain or loss of heat is called the *adiabatic curve,* and falls considerably below the hyperbola, which shows the law of expansion of air and other perfect gases.

Professor Zeuner takes an equation similar in form to Rankine's, but instead of the index $\frac{10}{9} = 1{\cdot}111$, he takes $1{\cdot}035 + \frac{x}{10}$, where x represents the dryness of the steam.

The equation is then,

$$p\ v^{\left(1\cdot035 + \frac{x}{10}\right)} = \text{constant.}$$

The smaller x is—that is, the more moisture the steam contains—the nearer will the expansion curve of the steam approximate to the hyperbola.

When the steam is quite dry $x = 1$, and the equation to the curve becomes

$$p\ v^{1\cdot135} = \text{constant.}$$

When the steam contains 30 per cent. of moisture $x = 0\cdot7$, and the equation becomes

$$p\ v^{1\cdot105} = \text{constant.}$$

Zeuner's equation holds good for all cases in which the steam does not contain more than 30 per cent. of moisture.

In an actual steam-cylinder, not fitted with a steam-jacket, but well clothed to prevent radiation of heat as much as possible, we have seen that a rather complicated action goes on, so that the form of the curve will probably be very different from this. Although in this case no heat is on the whole added to or taken from the steam, yet at the beginning heat is abstracted from the steam, and at the end heat is given out to it, the temperature of the cylinder being intermediate between the greatest and least temperatures of the steam.

The law according to which heat is exchanged between the sides of the cylinder and the steam is not accurately known, so that it is not possible to tell what the curve of expansion actually is. It is easily seen, however, that its general character would probably be somewhat of the following nature. Let A B C (Fig. 60) be the adiabatic curve, or the curve showing the expansion of steam without gain or loss of heat, and A B′ C′ the expansion curve of steam in an unjacketed cylinder. At first the sides of the

cylinder extract heat from the steam, and the pressure falls more rapidly than in the adiabatic curve, but at a certain point the sides of the cylinder begin to give out heat to the steam, and its pressure would probably rise above that of the adiabatic curve at the end of the expansion. Whether or not the final pressure may ever rise above the hyperbola, as it is sometimes supposed to do from the appearance of the indicator diagrams, cannot be answered positively until more is known of the action of the sides of the cylinders. It is most probable, however, that in an unjacketed cylinder at least it would not do so. It must not be forgotten that in an actual engine leakage of the

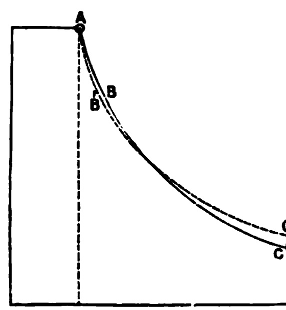


FIG. 60.

valves often occurs, and the effects of clearance may sometimes be to some extent overlooked. At high rates of expansion especially, these causes might tend to raise the pressure of the steam in the cylinder at the end of the stroke considerably above that due to the presumed rate of expansion, and lead to error in the determination of any definite law from the indicator diagrams.

For all practical purposes it will not be far wrong to assume that in unjacketed cylinders the pressure of the steam during expansion will vary inversely as its volume, so that when the volume is doubled the pressure is reduced to one-half, when it is trebled to one-third, and so on.

CHAPTER X.

METHODS OF INCREASING THE EXPANSIVE EFFICIENCY OF STEAM.

THE methods adopted for preventing the loss by liquefaction of steam in the cylinders of a steam-engine and thus increasing its expansive efficiency are :—

1. Surrounding the cylinder with a casing or jacket kept full of hot steam.
2. Superheating the steam before it is admitted to the cylinder.
3. Compound engines.

Steam-jacketing.—The steam-jacket was invented by James Watt, but it is not certain that he properly understood the principles of its action, but used it simply in furtherance of his idea of keeping the steam-vessel or cylinder warm. It certainly was not understood by his immediate successors, and most engineers at that period, arguing from the erroneous theory of caloric, which was then generally accepted, deemed it unnecessary and discontinued its use. Probably the difficulty of construction of jacketed cylinders in the then imperfect state of practical appliances and workmanship, as well as the want of knowledge of the true nature of the action and efficiency of the steam-jacket, contributed to its disuse, and it was considered to be sufficient to clothe the cylinders carefully with non-conducting materials to prevent loss from radiation.

The use of the steam-jacket was, however, retained in

a few special cases, such as the pumping engines for the Cornish mines, and in these the economy properly due to a high rate of expansion was realised. The covers of the cylinders of the Cornish engines were jacketed as well as the barrels, high rates of expansion were carried out in them, and these engines were for many years famous as being the most economical in the country.

In almost all other engines, and certainly in all marine engines, steam-jackets were not fitted. The result was that little or no practical advantage ensued if the steam were expanded more than from two to three times, and this became an article of faith amongst practical engineers in general, for its truth in the case of unjacketed cylinders was manifested not only by experiments made for the purpose of testing it, but also by every-day experience in the working of the engines.

When, however, the true nature of the action of expansion in a steam cylinder was discovered, and it was ascertained that the work of the engine was performed by the abstraction of heat from the steam and its conversion into mechanical work during expansion, which caused a portion of the steam to liquefy, it was found to be necessary, in order to increase the efficiency of the steam, to make provision for the addition of heat to the steam during its expansion.

Steam-jackets have, therefore, been re-introduced, and are fitted to nearly all modern engines, and it has been abundantly proved, both by experiment and by actual experience, that steam-jackets are necessary to enable high rates of expansion to be efficiently carried out and the greatest possible economy of heat attained.

The effect of the jacket is to prevent the changes of temperature of the metal of the cylinder and piston that take place in the unjacketed engine. The heat, added to the steam expanding in the cylinder, should be just sufficient

to prevent any appreciable quantity of it becoming liquid, so that the expansion diagram may be a curve representing the successive pressures and corresponding volumes of a given weight of saturated steam.

The relation between the pressures and volumes in the case of saturated steam is given by the equation:

$$p\, v^{\frac{17}{16}} = \text{constant},$$

so that the curve falls below the hyperbolic curve, which shows the expansion of a perfect gas, but above that of the expansion curve of steam expanding in a perfectly non-conducting cylinder, before discussed. The pressure, therefore, at the end of the expansion of steam in a jacketed cylinder, in which the heat supplied during expansion is just sufficient to keep the steam in a state of saturation, will be less than that due to the expansion of air or of a perfect gas, but greater than it would have been had the expansion taken place in a perfectly non-conducting cylinder, where the steam expands without either gain or loss of heat.

For one pound of saturated steam expanded in a jacketed cylinder, as above described, the value of the constant is about 475, so that the equation becomes:

$$p\, v^{\frac{17}{16}} = 475,$$

where p = absolute pressure in pounds per square inch and v = volume of one pound of steam in cubic feet.

The work done by the steam necessarily causes liquefaction to take place in jacketed as well as in unjacketed engines; but in the former case this liquefaction takes place in the jacket, where it produces no bad effect, and is simply discharged into the condenser, or hot-well, through the jacket drain cocks, to be returned again to the boiler; and no *waste* of heat ensues in consequence. It was

pointed out in the last chapter that the liquefaction due to the work done did not constitute in itself a loss of work, but that it was indirectly a loss of efficiency, from its action in equalising the initial and final temperatures; and, consequently, it is in the prevention of this action that the efficiency of the steam-jacket lies.

The foregoing is the theoretical action of the jacket, but it is not probable that in many cases it is so efficient as to entirely prevent the liquefaction in the cylinders. It is, however, certain that in all cases it tends to produce this effect, and the economy derived from its proper use is so decided as to be beyond the region of doubt.

Moisture in steam very considerably increases its power of conduction of heat. By means of the steam-jacket the steam is kept in a nearly dry state, so as to be a bad conductor of heat, and the moisture it contains, though probably sufficient to lubricate the piston, is not allowed to increase to such an extent as to carry away any considerable amount of heat from the metal of the cylinder and piston to the condenser.

Steam-jackets were at first fitted to the barrels of cylinders only; they were then added to the covers and ends, and in some cases arrangements have been made, by fitting hollow piston-rods and telescopic steam-pipes, to admit steam to the interior of the piston itself, thus causing the steam during its expansion to be entirely surrounded with a hot steam-jacket. There is, however, more advantage to be derived from jacketing the barrels than from jacketing the ends or the piston, because the friction of the piston keeps the surface of the cylinder-barrel comparatively clean, whilst the surfaces of the ends and piston soon become covered with a deposit which would prevent the passage of heat through them to the steam. The arrangements for admitting steam to the interior of the piston are also, necessarily, of a somewhat

complicated character, and the advantages derived have not been commensurate with the expense of first fitting, and the trouble and cost involved in keeping them efficient, so that these fittings are very rarely adopted. It is also easily seen that in an unjacketed cylinder the heating and cooling effects of the barrel would be more injurious than that of the ends and piston.

Many experiments have been made from time to time to ascertain the gain in economy due to the use of the steam jacket. More than twenty years ago the late Mr. John Penn made some experiments with steam-jackets. The working pressure was only 7 pounds per square inch, and when the cylinders were jacketed with steam of this pressure the gain was considerable and the economy increased as the pressure in the jacket was increased.

Some valuable quantitative experiments to ascertain the efficiency of steam-jackets were made by Mr. Emery on the cylinders of the United States survey vessel 'Bache,' referred to in Chapter IX. The results are given in the following tables. By comparing these trials it will be seen that the respective ratios of expansion, both with and without the jacket in use, are practically the same, so that the results may be taken to represent very fairly the economy due to the use of the steam-jacket.

Referring to the first six columns giving the results when the engine was worked as a simple expansion engine, we see that the most economical results were obtained, both with and without the jacket in use, when the ratio of expansion was about five times. When the jacket was not in operation, 26·25 pounds of feed-water were required per indicated horse-power per hour, whilst, when the jacket was used, only 23·15 pounds were required, showing in this case, when the simple engine was worked at its most economical point of cut-off, that a saving of 11·77 per cent. ensued by the application of the steam-jacket.

	Simple engines						Compound engines					
	Jacket in use			Jacket not in use			Jacket in use			Jacket not in use		
	1	2	3	4	5	6	7	8	9	10	11	12
1. Duration of trial hrs.	2·116	1·683	2·1	2·05	1·983	1·80	1·983	1·933	2·066	2·133	2·066	1·833
2. Pressure in boilers lbs.	79·5	81·07	80·83	78·11	79·62	81·0	80·125	30·214	80·33	80·31	80·28	82·0
3. Ratio of expansion . .	5·11	8·57	12·62	5·32	7·62	11·82	5·732	6·975	9·19	5·634	6·658	9·146
4. Vacuum in condensers, ins.	25·52	25·28	24·66	24·22	23·78	24·0	26·56	26·5	26·5	24·656	24·32	24·0
5. Revolutions per minute .	53·84	46·25	39·92	47·07	44·92	37·27	56·34	53·18	48·18	49·26	47·69	42·6
6. Indicated horse-power .	116·0	74·6	54·8	89·1	71·8	47·24	110·5	99·2	77·5	85·8	77·06	55·93
7. Water used per indicated horse-power per hour by measurement lbs.	23·15	24·09	27·11	26·25	29·62	35·07	20·37	20·33	20·71	23·21	23·036	23·765
8. Ditto estimated from indicator diagrams lbs.	16·25	15·58	16·42	17·35	17·76	21·03	15·25	14·77	15·76	12·346	12·312	12·699
9. Percentage of water used accounted for by indicator diagrams lbs.	70·2	64·66	60·55	66·11	59·95	59·95	74·89	72·65	76·07	53·19	53·45	53·44

At the higher rates of expansion the percentage of saving due to the jacket was much greater, being, when the rate of expansion was about eight times, as shown in columns 2 and 5, 18·67 per cent., and at the higher rates of expansion given in columns 3 and 6, 22·7 per cent. This is what might have been expected, for, as pointed out in Chapter IX., when condensation takes place at all in a steam cylinder, its amount progressively increases till the quantity of water condensed is as great as can be re-evaporated during the period of exhaust, and it tends to become a constant quantity, whatever the rate of expansion may be, so that the percentage of loss would increase with the rate of expansion, and consequently the higher the rate of expansion, the greater will be the resultant saving from the use of the steam-jacket.

When the engine was worked as a compound engine, the amount of feed-water required per indicated horse-power per hour was practically the same at all the rates of expansion given. The results are shown in columns 7 to 12 inclusive in the table, and it will be seen that the most economical results were obtained when the steam was expanded between six and seven times. The consumption of feed-water when the jacket was not used was 23·036 pounds per indicated horse-power per hour, whilst when the jacket was in operation, only 20·33 pounds were required per indicated horse-power per hour, showing in this case a saving of 11·75 per cent.

In discussing the case of liquefaction in the cylinders (Chapter IX.), it was pointed out that the percentage of heating and cooling surface was greater in small than in large engines, and that consequently when the cylinders were unjacketed, large cylinders were more economical per unit of power than small ones. By a similar course of reasoning, it will be seen that the jacket area is comparatively less in large than in small engines, for whilst

the volumes of the cylinders increase as the square of the diameter, the area of the jacket surface only increases directly as the diameter. The percentage of saving from the use of the jacket may, therefore, reasonably be expected to be greater in small than in large engines, and the effect of the addition of the jacket will be to render the expenditure of power per unit of area in large and in small engines more nearly equal to each other.

From this explanation of the action and efficiency of steam-jackets, it must be evident that the jacket should be kept filled with hot steam direct from the boilers. Some makers have caused the *exhaust* steam to pass through the jacket on its way to the condenser; but it is quite clear that comparatively cool steam cannot give out heat to steam of higher temperature, any more than water can flow up-hill, and this erroneous practice is now almost, if not entirely, abandoned. It is also objectionable to cause the steam, *on its way* to the cylinder, to pass through the jacket, because in this case it would be partially condensed before admission to the cylinder, and its initial pressure and efficiency consequently reduced.

Separate branch pipes should be led from the main steam-pipe direct to the jackets and the steam from the jackets should on no account be permitted to enter the cylinders. A drain pipe, fitted with a stop-cock and non-return valve, should connect the lowest part of the jackets with the hot-well or feed-tank; and the water condensed in the jackets should not be allowed to accumulate to any great extent, but should be blown into the hot-well or feed-tank, as the case may be.

All the theories relative to the action of the steam-jacket are based on the assumption that the heat supplied is only just sufficient to prevent any appreciable quantity of the steam in the cylinder from becoming liquefied, and not enough to superheat the steam at the end of the ex-

pansion. This assumption, as already stated, is founded on the fact, that dry steam is a very bad conductor of heat as compared with moist steam, so that when the steam has received a sufficient quantity of heat to make it dry, or nearly so, it will receive additional heat very slowly.

Superheated steam.— Another method employed to prevent liquefaction in the cylinder is to superheat the steam before its admission to the engines. It was pointed out in Chapter IV. that when the steam was kept in contact with the water from which it was generated, the temperature was dependent on the pressure, but when heat was added to the steam, in a separate chamber, the pressure being kept constant, the volume and temperature were increased, and the steam became superheated; that is, it received more heat than was necessary to keep it in a state of saturation.

The usual method of superheating steam in practice, is to cause it to pass, on its way to the engines, through or around tubes placed in the uptakes from the boilers, the opposite surfaces of the tubes being in contact with the hot gases escaping from the furnaces. Sketches of a superheater are given in Figs. 41 and 42. There can be little doubt that very much of the economy which has resulted from the use of superheaters has been due to the fact that, in addition to increasing the efficiency of the steam, they also increase the efficiency of the boiler by extracting some of the waste heat passing off into the atmosphere with the gases in the uptakes.

Sufficient exact data have not yet been obtained respecting the relations between the pressure, temperature, and density of superheated steam to enable a correct theory to be formed of its action; but as a provisional basis for calculation, it is generally assumed that it follows the law of perfect gases, and it is sometimes called *steam gas.* This assumption is, however, known to be inaccurate even

at temperatures of 30° to 50° above the temperature of saturation.

In boilers worked with steam pressures of 30 pounds per square inch and under, considerable gain in economy resulted from the use of superheated steam, for to steam of these pressures a considerable quantity of heat could be added without increasing the temperature to a dangerous extent. In any case, however, in which a superheater is applied it may be expected to increase the efficiency of the steam by preventing the use of wet steam or vapour, even if the steam be not superheated. Long steam-pipes, insufficiency of lagging or clothing, or slight but continuous priming in the boilers, tend to cause the steam to enter the cylinders in a super-saturated or moist condition, which reduces its efficiency by lowering the initial pressure, and, by its greater density, increasing the back pressure and impairing the vacuum. The superheater in this case would at least dry the steam, and the additional heat added to the steam would tend to prevent its falling below the temperature of saturation during its passage through the steam-pipes, &c.

In the early days of superheating, the late Mr. John Penn made a number of experiments on marine engines to which he had fitted superheating apparatus, in order to ascertain the economy due to the system, and found that a saving of fuel of about 20 per cent. was effected by superheating steam of 20 pounds pressure per square inch above the atmosphere to the extent of 100° Fahr. The temperature of saturated steam at this pressure is 259° Fahr., so that in these experiments the initial temperature of the steam was only 359° Fahr., which is not dangerously high. The increased economy in several ships of the General Steam Navigation Company, which were about the same time fitted with superheaters, appeared to corroborate Mr. Penn's experiments.

The working pressure in all these cases was about 20 pounds per square inch above the atmosphere. For higher pressures the gain resulting from superheating is much less, and for pressures of 60 pounds per square inch and upwards superheaters are not now fitted.

This is in consequence of the fact that when the temperature of the superheated steam is above a certain point, its effect on the engines has been found to be most injurious. The lubricants for the slides and cylinders become burnt up and hard, and act no longer as lubricants, so that the valves and pistons, working dry, grind the cylinder and slide-faces and very soon cause leakage. The packing in the stuffing-boxes also soon gets destroyed, and the working of the engine altogether becomes irregular and destructive. In some cases where superheated steam of too high temperature has been used, parts of the valve faces, valve rods, &c., when they were examined, appeared as if they had been put in a fire and burnt.

Sometimes the faces and backs of the slide valves and parts of the cylinder surfaces have been found coated with copper, as if they had been slightly electro-plated. It has been suggested that this action of highly superheated steam is, in all probability, due to the action of free hydrochloric acid resulting from the decomposition of the chlorides in sea-water by the great heat. The steam carries up into the superheater a portion of liquid water impregnated with the salts of sea-water, and these salts, deposited on the surfaces of the superheater and decomposed at high temperature, yield free hydrochloric acid. This first attacks the copper steam-pipes, forming chloride of copper, which, being carried over to the valve-chest, is in turn decomposed by contact with the iron, and the copper is deposited on the working surfaces.

It is only when superheated steam of very high tem-

perature is used that these injurious results follow; but they stringently restrict the extent to which superheating can be beneficially carried with the present system of lubrication. The limit of temperature appears to be practically about 360° to 370° Fahr. Above 400° Fahr. injury invariably ensues.

Steam of three atmospheres, superheated to 340° Fahr., falls to saturation when expanded in the ratio of about 3 to 2, or, more accurately, when cut off in the cylinder at 0·65 of the stroke. It will be seen from this how little is the margin for superheating with the pressures of seven to ten atmospheres now ordinarily used in marine boilers.

In some marine boilers worked with high-pressure steam, steam-driers are fitted. A sketch of this is given in Fig. 61, and it will be readily understood that, although the surface is not sufficient to superheat the steam to any great extent, it would at least have the effect of drying the steam by evaporating any particles of water carried over from the boilers, and this would have a beneficial effect on the working of the engines. The apparatus is fitted in the uptake, so that it is surrounded by the hot gases, which also pass up through the tube in the centre. A drain pipe is fitted at the bottom of the box.

One method that has been employed for superheating steam is to raise one portion of the steam to a high temperature, and then to cause this highly superheated steam to mix with the ordinary steam from the boiler before entering the cylinders. The result will be that the steam will enter the cylinder at a temperature intermediate between that due to its pressure and that of the steam in the superheater. The mixture thus made is often known as *combined steam.* The advantage of this course is, that it enables the degree of superheating to be easily and ac-

curately regulated so that its temperature on admission to the cylinders may not be injuriously high. In ordinary practice this is almost invariably done, as stated at p. 109; the valves leading to the superheater and those leading from the boiler direct to the engines being both opened, so that part of the steam is superheated and part not. By regulating the openings of these valves the steam may be

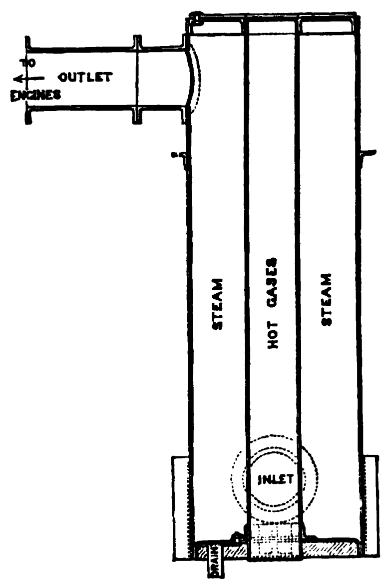


FIG. 61.

made to enter the cylinders at whatever temperature may be considered most desirable.

We pointed out on preceding page that when steam of a pressure of about 30 pounds per square inch above the atmosphere was superheated to the full extent desirable for ordinary engines, it fell to the point of saturation when expanded in the ratio of 3 to 2 only. It is evident, therefore, in this case, in order that a moderate amount of expansion

should be effectually carried out, steam-jackets should be fitted as well as superheaters. Most of the marine engines made to work with steam of 30 pounds pressure were therefore fitted both with steam-jackets and superheaters, and they formed the most economical type in use before the introduction of compound engines and high-pressure steam.

CHAPTER XI.

COMPOUND ENGINES.

THOUGH superheating and steam-jacketing tend to prevent liquefaction in the cylinders, they do not do so entirely in the quick moving engines now in use for marine purposes. In order, therefore, to realise the full benefits of a high ratio of expansion, the system of dividing the expansion into stages, carried out in two or more separate and successive cylinders, has been adopted. Engines of this description are generally called *compound engines*.

This system was invented by Hornblower as far back as 1781, and was shortly afterwards combined with Watt's condenser by Woolf. It was, however, soon abandoned, for it is obviously specially adapted for high pressures, which have only lately been brought into general use. Within the last few years it has been reintroduced, and the economy resulting from its use has been so decided that its application for marine purposes has become universal.

It is simply an extension of James Watt's principle of keeping the steam-vessel or cylinder as warm as possible and the condenser as cool as possible. With ordinary condensing engines the cylinder is open to the condenser for nearly the whole period of the return stroke of the piston, so that its temperature may be supposed to be cooled to nearly as great an extent as it was in the old days, when

the steam was actually condensed in the cylinder by the admission of a jet of water.

As we have before pointed out, the loss by liquefaction varies with the difference between the initial and final temperatures, so that with increased pressures and rates of expansion the loss from this cause would be proportionately increased. By dividing the expansion into two or more stages, therefore, this loss is reduced; for the cylinder into which the high-pressure steam is admitted is never opened to the condenser, and its temperature is never reduced below that due to the pressure in the intermediate receiver, say, from 12 to 20 pounds per square inch absolute. If any steam be condensed and re-evaporated in the first cylinder, the re-evaporated steam does useful work in the second cylinder, and the loss from liquefaction in the second cylinder is much reduced in consequence of the smaller range of temperature between admission and exhaust in that cylinder. The effective amount of clearance space in the cylinders of compound engines is also less than in simple engines working at the same power, with the same initial pressure and rate of expansion of steam, so that the loss of efficiency from this cause would also be decreased.

From investigations made both by the United States Government and by the British Admiralty it was clearly shown that the gain in economy by the substitution of the compound engine worked at 60 pounds pressure for the ordinary surface-condensing engine worked at 30 pounds pressure, was about 30 per cent.

It is not, however, quite fair to compare compound engines working with 60 pounds of steam with simple expansion engines working at 30 pounds pressure. We will therefore call attention to the results of some valuable experiments made to test the comparative efficiencies of the two systems, in which the boiler was the same in the two

cases, and the pressures, rates of expansion, and all the attendant circumstances, as nearly identical as possible.

The experiments were made by Mr. Emery, of New York, on the engines of the United States survey vessel 'Bache,' before alluded to, and the results are given in the following table. The steam-jacket on the large cylinder was in use on all the trials quoted. The small cylinder was not steam-jacketed, but was carefully felted and lagged to prevent loss from radiation.

On reference to the table it will be seen that in each case the consumption of water per indicated horse-power per hour was considerably less in the compound than in the simple expansion engine.

	Compound			Simple			
	1	2	3	4	5	6	7
1. Duration of trial, hrs.	1·983	2·066	1·733	2·116	1·683	2·10	1·883
2. Pressure in boilers, lbs.	80·125	80·333	80·375	79·5	81·071	80·333	30·875
3. Ratio of expansion	5·732	9·19	16·85	5·11	8·57	12·62	2·18
4. Vacuum in condenser ins.	26·56	26·5	24·5	25·52	25·29	24·66	24·0
5. Revolutions per minute	56·34	48·18	38·87	53·84	46·25	39·92	45·28
6. Indicated horse-power	110·5	77·5	46·4	116·0	74·6	54·8	66·7
7. Water used per indicated horse-power per hour by measurement lbs.	20·36	20·71	25·11	23·15	24·09	27·11	34·03
8. Ditto from indicator diagrams lbs.	15·25	15·76	18·53	16·25	15·58	16·42	24·04
9. Proportion of water used accounted for by indicator diagrams	·749	·761	·738	·702	·647	·606	·707

From columns 1 and 4 it is seen that with an expansion of between five and six times, which proved to be the most economical rate for each type of engine, the feed-water used per indicated horse-power per hour was in the simple engine 23·15 pounds, whilst in the compound engine it was only 20·36 pounds, showing in this case a gain in

economy by the use of the compound engine of rather over 12 per cent. At the higher rates of expansion the percentage of economy due to the compound engines was still higher.

The consumption of feed-water for the power developed was, both in the compound and in the simple engine, considerably greater at the high rates of expansion shown in columns 3 and 6 than at the lower rates. The non-compound engine was not tried at a higher rate of expansion than 12·62 times, whilst the compound engine was worked at a rate as high as 16·85 times. By increasing the ratio of expansion in the simple engine from 8·57 to 12·62 times, the expenditure of water was increased from 24·09 pounds to 27·11 pounds per hour. It is, therefore, probable that had an expansion of 16 or 17 times been carried out, the consumption of feed-water per indicated horse-power per hour would have been considerably over 30 pounds. In this case, therefore, the efficiency of the steam in the engine was considerably reduced as the rate of expansion was increased.

Though the actual efficiency of the steam is not so great at the higher rates of expansion, yet, as pointed out at p. 169, and as shown in columns 5, 6, and 7, it is advantageous when working at reduced powers to maintain a high initial pressure and employ high rates of expansion, instead of reducing the pressure and expansion.

High rates of expansion can be more economically and efficiently carried out in compound than in simple expansion engines, so it is clear that at reduced speeds the compound engine would be so much the more economical. This is of no small importance when we consider that it is only on special occasions that ships of war have to steam at full speed, and that during by far the greater period of their service they are steaming at speeds varying from 7 to 10 knots an hour.

Triple expansion engines.—For steam pressures above 100 pounds per square inch, which are now generally used, it has been found desirable to extend the compound system by dividing the expansion into three stages, so as to reduce the range of temperature in each cylinder. The results of experiments made to ascertain the consumption of coal in the S.S. 'Aberdeen' fitted with triple expansion engines by Messrs. R. Napier and Sons of Glasgow were communicated to the Institution of Naval Architects by Mr. A. C. Kirk in March 1882. The consumption of Welsh coal, ascertained during a four hours' full power run, was 1·28 pounds per I.H P. per hour, from which it was anticipated that the ordinary sea consumption would be from 1·5 to 1·6 pounds per I.H.P. per hour. This was stated by Mr. Kirk, in his remarks on Mr. A. E. Seaton's paper read at the Institution of Naval Architects in March 1885, to have been confirmed by the actual working of the ship and of her sister ship the 'Australasian' during several voyages to Australia and back. The ordinary seagoing consumption with two-cylinder compound engines may be taken to be about two pounds per I.H.P. per hour, so that the saving effected by the triple expansion engines in the cases quoted is just 20 per cent. Mr. Seaton in his paper gave some further experience with triple expansion engines. He quoted several interesting examples of similar ships performing similar voyages, some with two-cylinder compound engines, the others with triple expansion engines, and the saving effected by the triple expansion system was in each case stated to be about 20 per cent. The figures quoted by Mr. Seaton were taken from the records of coal-consumption kept by the shipowners.

Space forbids our dwelling any longer on these points, but we think, however, we have made it clear—

1. That the efficiency of the steam is increased by using it expansively within certain limits.

2. That in order to obtain the benefits of high rates of expansion, it is necessary either to superheat the steam or to surround the cylinder with a jacket filled with hot steam, or to use both these systems conjointly.

3. That high rates of expansion can be carried out most efficiently and economically in compound engines, in which the expansion is divided into two or more stages, so that the variations of pressure and temperature in each cylinder are comparatively small.

CHAPTER XII.

CONDENSATION OF STEAM.

BEFORE concluding this part of the subject, it will be necessary to refer to the way in which the *efficiency of the steam* has been increased by the introduction of the condenser.

The greatest possible efficiency of any heat engine depends on the difference between the initial and final temperatures at which it is worked, and is represented by the formula—

$$\text{Maximum efficiency} = \frac{T_1 - T_2}{T_1 + 461}$$

where T_1 = initial temperature.
T_2 = final temperature.

In the non-condensing engine (often called by the misleading term 'high-pressure engine') in which the steam, after having done its work in the cylinder, escapes to the atmosphere, the back pressure is not less than from 3 to 4 pounds per square inch above the atmosphere, and the lower limit of temperature is that corresponding to an absolute pressure of from 18 to 19 pounds per square inch, say about 224° Fahr. When a condenser is used so that a partial vacuum is formed in the cylinder behind the piston, the back pressure is only 3 to 4 pounds absolute, so that the lower limit of temperature in this case is only about 150° Fahr. By the introduction of the condenser, therefore, the difference between T_1 and T_2 is considerably

increased, and consequently the efficiency of the steam is increased likewise. It is clear that the lower the initial pressure of the steam is, the greater the increased efficiency due to the use of the condenser will be, for in this case the fact of having a low back pressure is of considerably greater importance.

If we imagine two engines, one condensing, the other non-condensing, working with steam of 60 pounds per square inch above the atmosphere, or 75 pounds absolute, for which the temperature is about 307° Fahr., the non-condensing engine would be working between the limits 307° and 224°, whilst the limits in the case of the condensing engine would be 307° and 150°. The relative maximum efficiency of the condensing to that of the non-condensing engine would therefore be:

$$\frac{307-150}{307-224}=\frac{157}{83}$$

or nearly two to one. That is, the efficiency of the steam could be nearly doubled by the addition of a suitable condenser.

This may also be illustrated by means of an indicator diagram. Suppose that the initial pressure and rate of expansion are the same in each case. Then the steam line, until the end of the forward stroke, would be the same both in the condensing and non-condensing engines, the only difference in the two cases being in the position of the line of back pressure.

In the non-condensing engine the back pressure line D E, Fig. 62, will be about 3 or 4 pounds above the atmospheric line H K. In the condensing engine the pressure at the end of the stroke falls considerably below the atmospheric pressure, and the back pressure will only be 3 to 4 pounds absolute, or, say, 11 to 12 pounds below the atmospheric line, as shown by the line F G, so that by the

application of the condenser the work done by the same weight of steam is increased by an amount represented by the area E D F G.

The forward pressure would be the same in each case, but in the non-condensing engine this would be resisted by a back pressure of 18 to 21 pounds per square inch; whereas with the condenser, the pressure resisting the forward motion would be only 3 to 4 pounds absolute. It is clear that in a non-condensing engine the cut-off should never be early enough to cause the pressure of steam to fall below the atmosphere before the completion of the

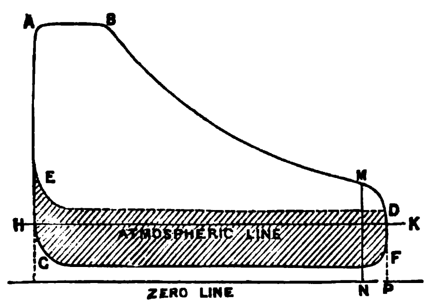


FIG. 62.

stroke; for this would necessitate the latter part of the stroke being performed by the expenditure of the work accumulated in the fly-wheel during the earlier part of the stroke, and would probably cause difficulty in starting, and prevent smoothness and regularity in working.

It is evident that an equal quantity of steam would be used by both engines, because the absolute pressure of the steam N M just before release, which represents the steam used, is the same both in the condensing and the non-condensing diagrams. It is important to remember that in all calculations respecting steam, *absolute* and not *relative* pressures must be used.

Professor Cotterill, in his 'Notes on the Theory of the Steam Engine,' published in 1871, calculates for a few

cases the number of pounds of steam actually required per hour if the engine were perfect, both in the case of a condensing and of a non-condensing engine. The results are given in the following table, and clearly show the gain in efficiency due to the condensation, and also that the percentage of gain is greater at low than at high pressures.

Pounds of steam per indicated horse-power per hour.

Initial pressure in atmospheres	Condensing	Non-condensing	T_1
	lbs.	lbs.	
2	11·2	50·4	249°
4	9·2	24·8	291°
6	8·3	18·9	318°
8	7·7	15·9	340°

In the condensing engine it was assumed that the feed-water was taken from the condenser at a temperature of 100° Fahr., and in the non-condensing engine that the exhaust steam had been used to raise the temperature of the feed-water to about 212° Fahr.

The condensers at first used were known as jet injection condensers. In these the steam, after leaving the cylinders, enters the condensing chamber, into which a jet of cold water is injected, which condenses the steam by actual contact and mixing. The orifice of the injection pipe is generally covered with a rose or perforated plate to distribute the water and facilitate its mixing with the steam from the cylinder. Sketches of the condenser, with a description of its construction and details, are given in Chapter XVII.

In this form of condenser, the condensed water, vapour, air, &c., from the condensing chamber are pumped by the air-pump into another chamber called the hot-well, which, when the engines are at work, is in connection with the sea by means of the discharge or waste-water valve on the ship's side. This valve should always be self-acting, so

that in case of the engines being started before the valve is opened, the condenser may not be burst by excessive pressure.

The suction valves to the air-pump are generally called foot valves, and the exit valves the head or delivery valves. The *head valves* should be placed above the highest part of the pump chamber, so as to insure the air and vapour being expelled each stroke. Pockets and spaces in the pump chamber where air might collect should also be carefully avoided. If any accumulation of air exist in the pump chamber, the valves will not work properly, and a good vacuum cannot be obtained. Arrangements should also be made to insure the valves being always covered with water when the engines are at work.

Vertical air-pumps, especially when single-acting, and fitted with bucket plungers, are more efficient than horizontal pumps with solid plungers. For the action of the air-pump it is essential that the vacuum in the pump chamber should be greater than that in the condenser, as the foot valves are only worked by the difference between the absolute pressures in these two chambers, and, with the vertical pump, the bucket dips into the water over the foot valves, the air and vapour, being on top of the water, passing through first.

From the hot-wells the feed pumps draw their supply for the boilers, a cock or valve being fitted on the suction orifice in the hot-well to enable the supply to be regulated.

The quantity of water required to condense the steam in a jet condenser is nearly thirty times the amount of steam condensed. This may easily be seen from the following calculation:

Suppose the steam entering the condenser to have a pressure of 3 to 4 pounds absolute, corresponding to a temperature of about 150° Fahr. The temperature of the condensed water in the hot-well may be assumed to be

100° Fahr., which is about the best working temperature for the condenser. The temperature of the sea-water used for injection is taken at 60° Fahr.

Then each pound of steam entering the condenser at a temperature of 150° Fahr. is finally reduced to water at 100° Fahr.

The latent heat of one pound of steam at 150° Fahr. is—

$$=966-0{\cdot}7(150-211)=1010 \text{ thermal units.}$$

In addition to this, the sensible heat is reduced from 150° to 100°, which represents fifty thermal units. Therefore the total heat given out by the condensation of the pound of steam is $=1010+50=1060$ thermal units.

This is all expended in heating the injection water, the temperature of which has been raised from 60° Fahr. to 100° Fahr. during the operation, which represents 40 thermal units for each pound used; and the amount of heat given out by the steam is clearly equal to that taken up by the injection water.

If N = number of pounds of injection water required per pound of steam, the total heat absorbed by the water during the condensation will be $N \times 40$ thermal units.

Consequently,

$$N \times 40 = 1060$$

$$\text{and } N=\frac{1060}{40} = 26{\cdot}5 \text{ lbs.}$$

Of course the higher the temperature of the condensing water, the greater will be the quantity required, so that more injection water is necessary in the tropics than in colder climates.

In consequence of the large quantity of injection water required, it is evident that the feed-water for boilers of ships supplied from jet injection condensers is practically as salt as sea-water itself. This necessitates the frequent blowing out of a portion of the water in the boilers to prevent undue incrustation on the heating surfaces, and

the danger that would probably result from it, as on the evaporation of sea-water the solid constituents are left behind in the boiler.

This operation entails a considerable waste of heat, for the water blown out has been raised in temperature from 100° Fahr. to, say, from 260° to 300°, and the heat expended in doing this will obviously be wasted.

Sea-water contains about $\frac{1}{33}$rd part, by weight, of solid matter in its composition, and the proportion necessary to be blown out to keep the water at any particular density may be easily calculated.

Let x = quantity of water evaporated

y = „ „ blown out

Then $x + y$ = the quantity required to be admitted as feed.

Suppose the water in the boiler to be kept at a constant density equal to 'n' times that of sea-water, then the quantity of solid matter blown out of the boiler is

$$= y \times \frac{n}{33}.$$

The quantity of solid matter pumped into the boiler with the feed-water during the same period is represented by

$$(x + y) \times \frac{1}{33};$$

and since the density of the water in the boiler is kept constant, the solid matter blown out must be equal to that pumped in.

Therefore

$$(x + y) \frac{1}{33} = y \cdot \frac{n}{33}$$

$$x + y = n . y$$

or,

$$\frac{y}{x} = \frac{1}{n-1}.$$

For example, suppose the density in the boiler to be kept at three times that of sea-water, then

$$n = 3, \text{ and } \frac{y}{x} = \frac{1}{2} \text{ or, } y = \frac{1}{2}x;$$

that is to say, the quantity necessary to be blown out to keep the density in the boiler constant at three times the density of sea-water, or at 30 degrees by the Service hydrometer, is one-half the quantity of water turned into steam to be used in the engine.

If the density be kept at 20°, or twice the density of sea-water,

$$n = 2, \text{ and } \frac{y}{x} = 1 \text{ or } x = y.$$

In this case the quantity blown out must be equal to that evaporated.

If the density be kept at 15 degrees, $n = 1\frac{1}{2}$, and

$$\frac{y}{x} = 2 \text{ or } y = 2x;$$

that is to say, that in order to keep the density of the water in the boiler at one and a half times that of sea-water, it would be necessary to blow out twice as much water as that evaporated, so that for every pound of steam used in the engine 3 pounds of feed-water would be required to be supplied to the boilers.

We will now calculate the percentage of heat wasted by blowing out under these circumstances.

Let T_1 = temperature of the water in the boiler,
T_2 = ,, the feed-water.

The quantity of heat that must be added to one pound of water at temperature T_2, in order to turn it into steam at temperature T_1, is given, in thermal units, by the formula

$$H_1 = (T_1 - T_2) + 966 - 0{\cdot}7\,(T_1 - 212).$$

Therefore, the heat necessary to evaporate x pounds of water

$$= xH_1 = x\{(T_1 - T_2) + 966 - 0{\cdot}7\,(T_1 - 212)\}.$$

Each pound of water blown out has been raised in tem-

perature from T_2 to T_1, and has consequently had added to it, $(T_1 - T_2)$ thermal units, so that the total amount of heat wasted by blowing out is represented by

$$y\,(T_1 - T_2) \text{ thermal units.}$$

The total amount of heat expended on the $(x + y)$ pounds of feed-water admitted to the boiler is therefore

$$= x\{(T_1 - T_2) + 966 - 0{\cdot}7\,(T_1 - 212)\} + y(T_1 - T_2)$$
$$= (x + y)\,(T_1 - T_2) + x\{966 - 0{\cdot}7(T_1 - 212)\}$$

The percentage of heat wasted is therefore equal to

$$\frac{y\,(T_1 - T_2) \times 100}{(x + y)\,(T_1 - T_2) + x\{966 - 0{\cdot}7\,(T_1 - 212)\}.}$$

Suppose the working pressure of steam in the boiler to be 30 pounds per square inch, the density to be kept at 15 degrees, or one and a half times that of sea-water. Temperature of feed-water, $T_2 = 100°$ Fahr. In this case the quantity of water blown out is twice that evaporated, or $y = 2x$, so that for every 3 pounds of feed-water admitted to the boiler, one is evaporated and two blown out.

The temperature of the water in the boiler, $T_1 = 275°$.

Heat expended in evaporating one pound of the feed-water:

$$\begin{aligned} H_1 &= T_1 - T_2 + 966 - 0{\cdot}7\,(T_1 - 212) \\ &= 275 - 100 + 966 - 0{\cdot}7\,(275 - 212) \\ &= 175 + 966 - 0{\cdot}7 \times 63 \\ &= 175 + 966 - 44 = 1097 \text{ thermal units.} \end{aligned}$$

The heat wasted on the two pounds blown out:

$$\begin{aligned} &= 2\,(T_1 - T_2) = 2\,(275 - 100) \\ &= 2 \times 175 = 350 \text{ thermal units.} \end{aligned}$$

Total expenditure of heat on 3 pounds of feed-water:

$$= 1097 + 350 = 1447 \text{ thermal units.}$$

Therefore the percentage of heat wasted is equal to

$$\frac{350 \times 100}{1447} = 24{\cdot}2 \text{ per cent.}$$

If the density be kept at 20 degrees, or twice that of sea-water, the quantity of water blown out is equal to that evaporated.

By applying a similar mode of calculation we find:

Heat expended in evaporation	=1097	thermal	units.
Heat wasted in blowing out	= 175	„	„
Total heat expended	= 1272	„	„

Percentage of waste of heat:

$$=\frac{175\times 100}{1272}=13{\cdot}76 \text{ per cent.}$$

If the density be kept at 30 degrees, or three times that of sea-water, the quantity of water blown out is only one-half that evaporated.

In this case:

Heat expended in evaporation	= 1097	thermal	units.
Heat wasted in blowing out	= 87·5	„	„
Total heat expended	= 1184·5	„	„

Percentage of waste of heat:

$$=\frac{87{\cdot}5\times 100}{1184{\cdot}5}=7{\cdot}4 \text{ per cent.}$$

Formerly the density of the water in boilers of this class was rarely allowed to much exceed 15 degrees by the hydrometer, at which density we have seen that more than 24 per cent. of the total heat expended on the water was necessarily wasted by blowing off. It has, however, been found by experiment that the water in boilers fed with sea-water may be kept at a density of from 25 to 30 degrees, without any injurious results. In fact, in most cases the boilers have been found to be in a better condition, with respect to scale, &c., by working at the higher densities, and the calculations just given clearly show the considerable gain in

economy that may result from this method of working. The highest density of water now allowed for boilers of this class is 30 degrees, or three times the density of sea-water.

The use of the jet condenser retarded for many years the use of high-pressure steam, as it was considered to be both injurious and dangerous to work boilers at high pressures if fed constantly with sea-water, the solid matter being more freely deposited at high temperatures, and the results of the deposit would be more serious.

Surface condensation.—This led to the introduction of what is known as surface condensation, and it is owing to the application of this system that the use of high-pressure steam has become so general for marine purposes in recent years. It is not, perhaps, too much to say that the great advance in the economy of working of marine engines during the past few years has been mainly due to the introduction of service condensation, which has rendered the employment of high pressures with high rates of expansion of steam practically possible, by which the efficiency of the steam in the engine has been so greatly increased.

In surface condensation the steam is not condensed by actual mixing with cold water, but, as the term implies, by contact with a cold surface. This usually consists of the surfaces of a number of tubes, kept cool by a stream of water on the side opposite to that on which the steam impinges. Sketches of surface condensers are given in Chapter XVII.

The condensed water, therefore, does not become mixed with the sea-water, but is kept separate, so that the boilers can be supplied with pure, or nearly pure, water. The loss from blowing out can by this means be prevented, and as the feed-water is practically free from solid matter, very high pressures may be used without fear of

injury to the boiler from incrustation These advantages are very tangible, and have led to the universal adoption of the surface condenser for marine engines.

The advantages of surface condensation for marine engines may be briefly stated as follows:—It renders the condition of the condensing water of no importance, so that whether it is salt, muddy, acid, or otherwise impure, the boiler may be supplied with pure water condensed from the steam without contact with the impure water in which the ship floats, except the comparatively small quantity required to compensate for unavoidable losses; some ships carry supplies of fresh water for this purpose. High pressures of steam may consequently be used without any fear of injurious incrustation or deposit, and the efficiency of the steam may be increased by the employment of higher rates of expansion, &c.

A much larger quantity of water is required for condensation with surface than with the common jet condensers. Suppose the circulating water to be raised in temperature to the extent of 15 degrees by passing through the condenser. As we saw in the case of the jet condenser, the heat given out by each pound of steam condensed is 1060 thermal units.

If N be the number of pounds of circulating water required per pound of steam used, we have:

$$N \times 15 = 1060.$$

$$\therefore \quad N = \frac{1060}{15} = 71 \text{ lbs. nearly.}$$

So that 71 pounds of circulating water are required per pound of steam condensed.

This water is either forced or drawn through the condenser by a reciprocating pump worked off the engine itself, or more generally by a separate centrifugal pump, the details of which are described in Chapter XVII.

PART IV.

THE MECHANISM.

CHAPTER XIII.

REGULATING AND EXPANSION VALVES AND GEAR.

WE will now trace the action of the steam through the engine, describing and explaining the uses of the various parts of the mechanism as we proceed.

Regulating valve.—The steam, after leaving the separator, proceeds along the remainder of the steam-pipe, till it arrives at the regulating valve for the engines, still often called the throttle valve, which was the form of valve originally used for the purpose. The object of this fitting is to provide means for regulating the supply of steam to the engines, so that the speed may be varied as required.

Throttle valve.—This was the original regulating valve, and consisted simply of a flat plate or disc in the pipe, having a central spindle passing through to the outside of the pipe, by means of which it could be turned, so as to either close or open the passage as required. A section of a throttle valve is shown in Fig. 63. It was first introduced by James Watt, but is now seldom fitted, as it is difficult to keep a valve of this description tight at present pressures of steam, and it does not admit of sufficiently

exact regulation of the engines for the requirements of modern practice.

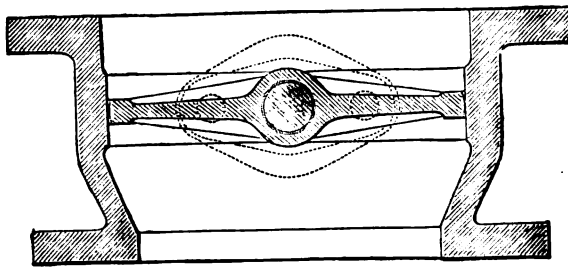


FIG. 63.

It has been generally superseded by two kinds of regulating valves, viz., the gridiron valve and the conical

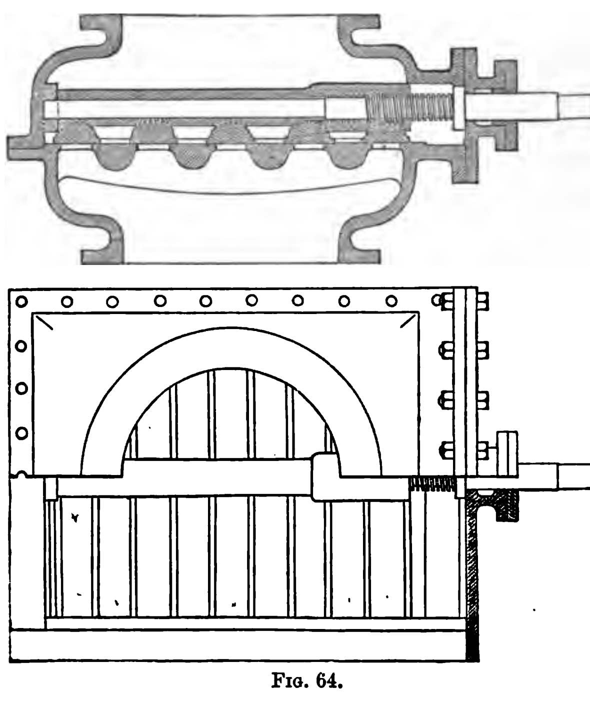


FIG. 64.

valve, the latter being generally double-seated, so as to be nearly in equilibrium.

Gridiron valve.—This is a flat valve, consisting of a number of bars with open spaces between them. The valve slides on a seating with a corresponding number of ports, so that a comparatively large area is opened for the passage of steam, by a small movement of the valve.

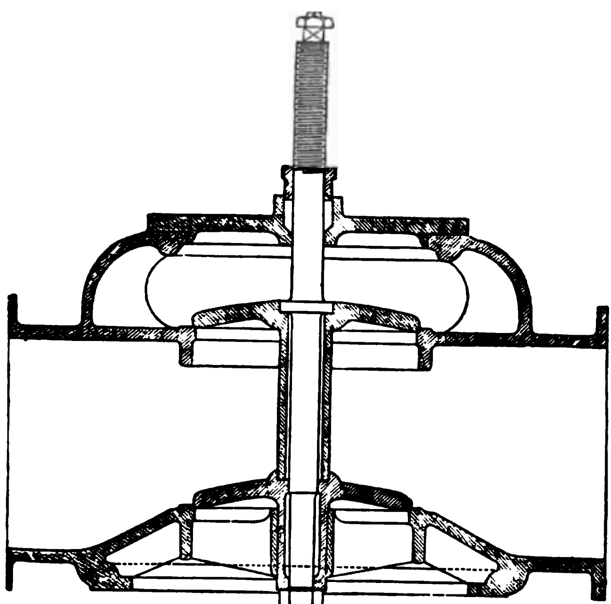


FIG. 65.

This kind of valve was much used in marine engineering, especially in cases where the steam pressure did not exceed 30 pounds per square inch. Its general form and arrangement are shown in Fig. 64. In some cases the valve was made hollow, with two faces, so that the steam flowing on both sides kept the valve in equilibrium.

Double-beat valve.—One form of the double-beat regu-

lating valve is shown in Fig. 65. It consists of two conical valves on the same spindle; the steam pressure acts on the top of one valve and on the bottom of the other, so that the valve is nearly in equilibrium, and little force is required to move it from its seat, the only resistance to be overcome being the difference of the pressures on the two valves. The largest diameter of the lower valve in this arrangement must obviously be somewhat less than the smallest diameter of the upper valve, to enable the valve to be put in its place. The expansion of the spindle, when hot, tends to alter the distances of the two valves from each other, which may cause leakage when under steam. The amount of opening for a certain height of lift is practically double that for the same lift of an ordinary single conical valve of the same diameter.

Cornish double-beat valve.—Another kind of double-beat valve, generally known as the Cornish valve, is shown in Fig. 66. This is probably the best form of valve for enabling a large passage to be opened and closed for the flow of steam, or any other fluid, under a high pressure. In this arrangement the valve-seatings are fixed in the centre of the valve-box, and the valve consists of a cylindrical casing, of the section shown in Fig. 66, having two faces corresponding with the valve-seatings. When the valve is raised the steam passes the two seatings, and from its construction it is evident that it can be arranged to be in perfect equilibrium at all pressures.

These regulating valves are generally worked by screw gearing, so that the speed of the engines may be adjusted with great exactness. Screw gear is preferable to levers, as there is less backlash, and the gear should be as direct and simple as possible.

Expansion valve.—After passing the regulating valve, the steam either enters the expansion valve box or the slide casing direct, according to the construction of the engines.

Modern engines work at such high rates of expansion, that it is often desirable to fit separate valves, usually called expansion valves, for the purpose of cutting off the admission of the steam at a sufficiently early part of the stroke. The sole office of these valves is to stop the ad-

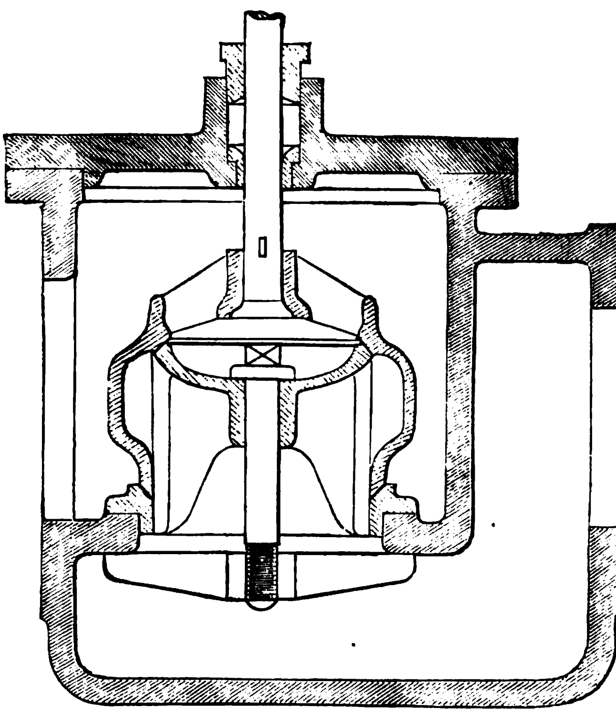


FIG. 66.

mission of steam at the required point, and they have nothing further to do with the distribution of the steam, which is done by the slide-valve.

Throttle expansion valve.—In the earlier engines which were fitted for expansive working, an ordinary throttle valve, actuated by cams on the crank-shaft through a

series of levers, was used for cutting off the admission of the steam.

A *cam* is a curved plate or groove, which communicates motion to another piece of the mechanism by the action of its curved edge.

The general arrangement of this description of expansion gear is shown in Fig. 67. When the roller is on the circular part of the cam, the throttle valve is open, but when the roller is raised by the projections on the cam, as

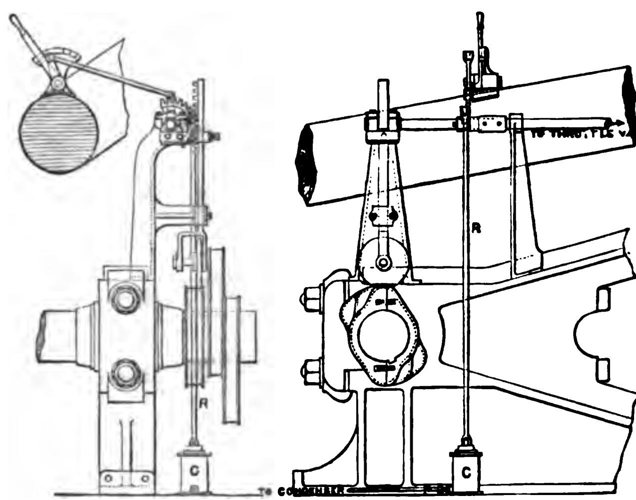


Fig. 67.

the shaft revolves, it acts on the intermediate levers and shafts and closes the throttle valve, thus cutting off the admission of steam to the cylinder. The different grades of expansion are obtained by a suitable arrangement of the steps of the cam, the roller being fitted so that it may be moved by hand to the proper step for the rate of expansion required. This gear was usually fitted for three grades of expansion only, which was found to be sufficient for the purpose, and any greater number would involve

increased width in the cams, which would be inconvenient practically.

Vacuum pump.—The rollers were at first kept in contact with the cams by means of springs, but it was found to be more efficient to connect the lever carrying the rollers by means of a rod, R, to a plunger or piston working air and steam tight in a small cylinder, C, as shown in Fig. 67. This arrangement was technically called the vacuum pump. The upper part of this cylinder was open to the atmosphere, but the lower end was connected to the condenser by means of a small pipe. When the gear was at work the roller was kept constantly pressed against the cam

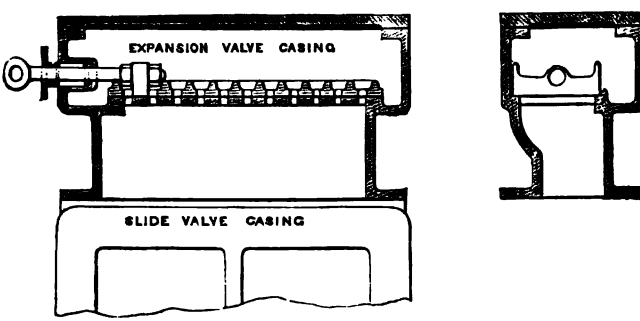


FIG. 68.

by a force equal to the difference between the atmospheric and condenser pressures, say 10 to 12 pounds per square inch, multiplied by the area of the piston of the apparatus.

Expansion valves as now fitted are generally sliding valves, and work either on separate seatings, placed usually at the tops of the slide-jackets, or directly on the backs of the main slide-valves themselves.

Gridiron expansion valve.—When the expansion valve works on an independent seating it is usually of the gridiron type, and its general form and arrangement are shown in Fig. 68. The steam from the steam-pipe enters the expansion valve casing, and when the valve is in such a

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A *cam* is a curved plate or groove, which communicates motion to another piece of the mechanism by the action of its curved edge.

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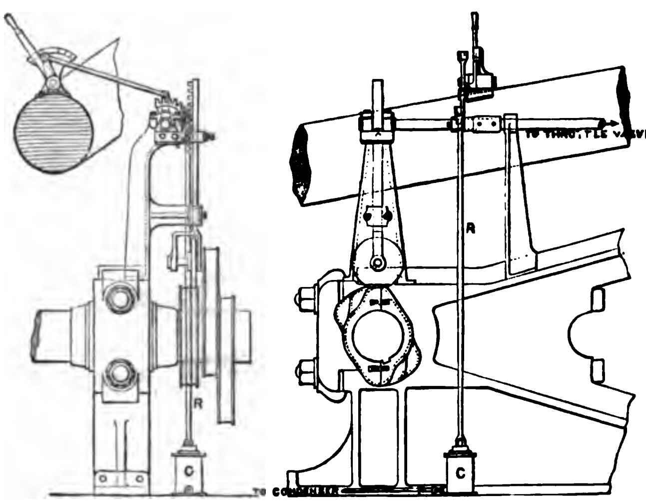

Fig. 67.

the shaft revolves, it acts on the intermediate levers and shafts and closes the throttle valve, thus cutting off the admission of steam to the cylinder. The different grades of expansion are obtained by a suitable arrangement of the steps of the cam, the roller being fitted so that it may be moved by hand to the proper step for the rate of expansion required. This gear was usually fitted for three grades of expansion only, which was found to be sufficient for the purpose, and any greater number would involve

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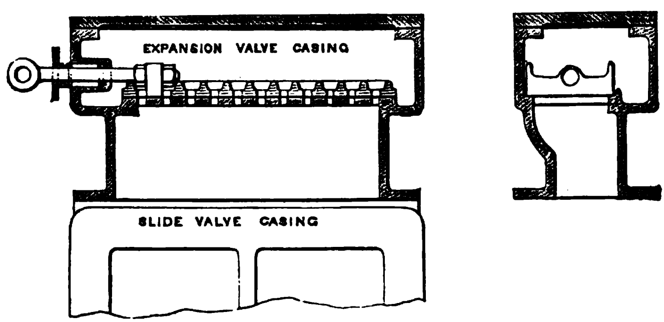

FIG. 68.

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Expansion valves as now fitted are generally sliding valves, and work either on separate seatings, placed usually at the tops of the slide-jackets, or directly on the backs of the main slide-valves themselves.

Gridiron expansion valve.—When the expansion valve works on an independent seating it is usually of the gridiron type, and its general form and arrangement are shown in Fig. 68. The steam from the steam-pipe enters the expansion valve casing, and when the valve is in such a

position that the ports are open, the steam passes into the slide-valve casing underneath. When the valve is in its middle position all the ports are wide open. A great number of ports are desirable, so that the amount of opening may be considerably affected by a small motion of the valve, and the cut-off, therefore, sharper, and more effective.

This arrangement for expansion is not very efficient for early points of cut-off, on account of the very large clearance space between the valve and the piston that has to be filled with fresh steam at each stroke. When the admission of steam is cut off by the expansion valve, the whole of the slide-valve casing is full of steam, which enters the cylinder, and causes the actual degree of expansion to be much less than that given by the point of cut-off of the valve itself. This point is further discussed in Chapter XXIV. It is not of so much importance in compound as in simple expansion engines, as in the former case the ratio of the cylinders themselves provides for a considerable amount of expansion.

These expansion valves are worked by eccentrics on the crank-shaft, and the extent of the travel of the valve is regulated by a link, fitted with a sliding block attached to the expansion-valve rod. The gear should be so arranged that when the engines are not being worked expansively, the motion of the valve should, if possible, be reduced to zero, the valve remaining in its central position, with all the ports wide open. In the majority of arrangements this is not exactly accomplished, though the motion of the valve is reduced to such an extent that it has no appreciable effect on the passage of the steam. In these cases the width of the bars in the seating is usually made greater than that of the bars in the valve, to allow for the minimum motion of the valve without decreasing the area for admission of the steam when working at full power. By

altering the position of the block in the link and so increasing the travel of the valve, the point of cut-off is made earlier. The greater the travel the higher will be the rate of expansion.

Fig. 69 shows the general arrangement of expansion gear for valves of this description, and the method of its action can be easily seen from the diagram. C is the

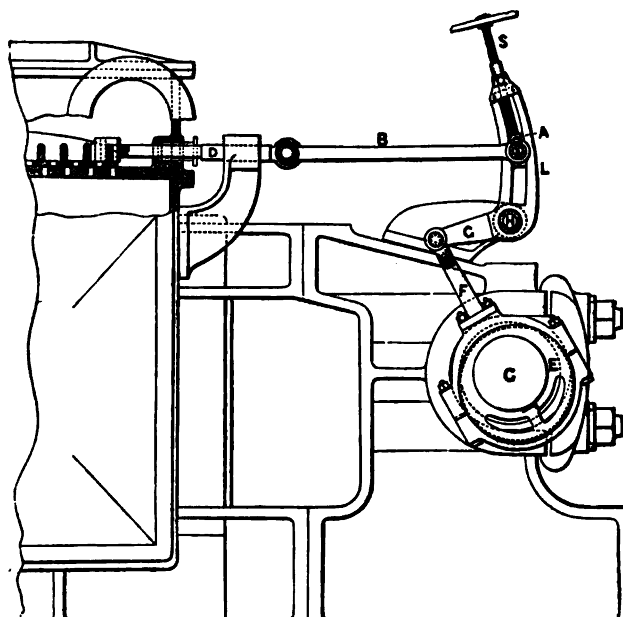


FIG. 69.

crank-shaft on which the eccentric, E, is keyed. The eccentric rod F is attached to a lever, G, forming the lower arm of the link L, which it causes to vibrate on the joint or fulcrum H. In the long arm of the link there is a slot in which a block, A, slides, which, by means of a connecting rod, B, works the expansion valve rod D, and consequently the valve itself to which the rod is attached. It

will be readily seen that by altering the position of the block A by the screw S, the travel of the valve is altered, and the point of cut-off of the steam regulated as may be required.

When these expansion valves are set so as to cut off the steam at an early point, they always open again before the piston arrives at the end of the stroke. If this should occur before the steam port has been closed by the slide-valve, there will be a second admission of steam, which would be shown on the indicator diagram by an elevation at the lower end of the expansion curve, and would cause a waste of steam. The slide-valve should therefore be so arranged as to cut off the steam before this second opening of the expansion valve. In any case in which the second admission does occur, it may be prevented, and the injurious effect obviated, by 'linking up,' to cause the slide-valve to cut off the steam at an earlier part of the stroke.

Expansion valves on the backs of main slide-valves.—The expansion valves that work directly on the backs of the main slide-valves are of two descriptions:

1. Gridiron valves similar to those just described, in which the point of cut-off is regulated by the amount of travel of the valve.

2. Valves constructed of two separate but similar parts, connected together by a right and left-handed screw, by means of which their distance from each other may be varied. In this arrangement the travel of the valve is constant, and the point of cut-off is regulated by the distance between the two plates that form the valve, which can be varied as required.

Sketches of these valves are given in Figs. 70 and 71. So far as the distribution of the steam is concerned these valves, which work directly on the backs of the main slides, have a great advantage over the gridiron valve, shown in

Fig. 68, as the clearance space that has to be filled with steam each stroke is much diminished. It has, however, in many cases been found difficult to properly lubricate the working surfaces, especially for large engines, and the valves have worn away rapidly, causing excessive strains on the gear from the great friction.

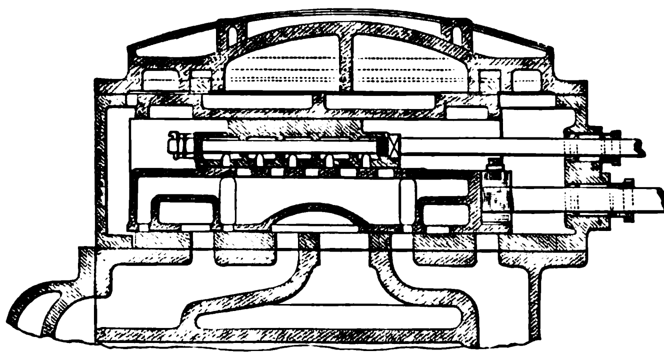


FIG. 70.

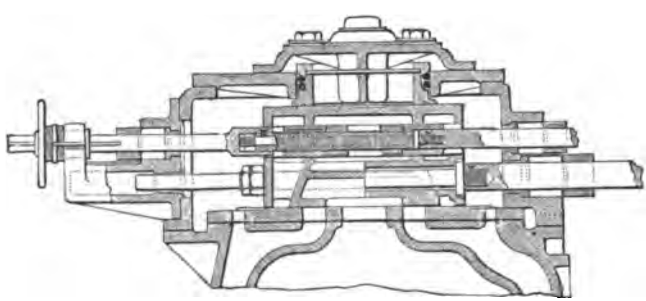


FIG. 71.

For marine engines generally, if proper precaution be taken in the design to reduce the clearance spaces to as great an extent as possible, the ordinary gridiron expansion valve on the slide-casing will be found to be more efficient in the long run than the other type, working directly on the back of the slide-valve. Though theoreti-

cally less perfect as cut-off valves than those on the backs of the slide-valves, there is no difficulty in keeping them in good working order, and no fear of causing excessive pressure and grinding on the slide and expansion valve faces, with the consequent serious straining of the valve gear, which have so frequently followed the use of the otherwise more perfect valves.

In the more recent compound engines, and in triple expansion engines, in which the cylinders themselves pro-

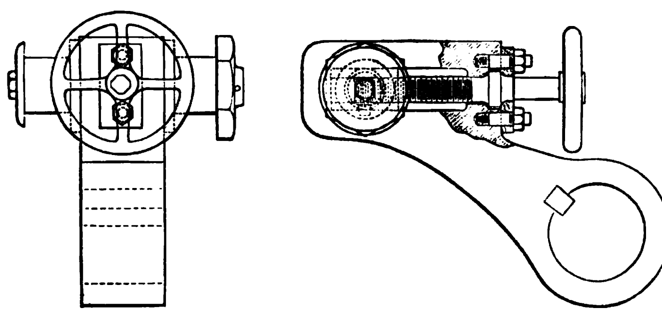


FIG. 72.

vide for a considerable range of expansion, separate expansion valves have been dispensed with. The earlier points of cut-off required when working at reduced power are obtained by the use of the link, and in order to distribute the work between the respective cylinders, the reversing arms are fitted with sliding pins to which the suspending rods are attached, so that the degree of expansion in each cylinder may be regulated as required. This arrangement is shown in Fig. 72.

CHAPTER XIV.

SLIDE-VALVES AND FITTINGS.

Slide jacket.—The steam, after passing the expansion valve, enters the slide jacket or casing, which is simply a rectangular or cylindrical box bolted to the cylinder, in which the slide-valve works.

Slide-valve.—The distribution of steam in the cylinder of a steam-engine, involving the processes of admission from the boiler, expansion, and finally exhaust into the condenser, is now usually effected by the agency of a single valve, called the slide-valve. The slide-valve is one of the most important parts of the engine, and on the skill and care exercised in its design and fitting, the efficient and satisfactory working of the machinery will greatly depend.

A section of an ordinary single-ported slide-valve is shown in Fig. 73.

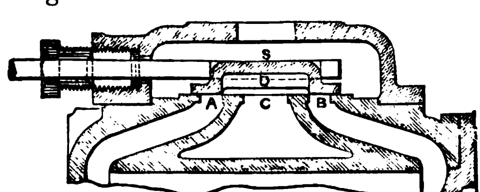


FIG. 73.

In the cylinder face there are three ports, marked respectively A, B, and C. A and B are the steam ports, one leading to each end of the cylinder, and C is the exhaust

port leading to the condenser. S is the slide-valve, which is rectangular in plan, and the hollow space, D, in the centre, is called the exhaust cavity of the valve. The valve is shown in its central position, and it will be seen that it not only closes the steam ports, but overlaps the edges for some distance. The object of this will be explained further on.

The slide-valve works in a steam-tight casing to which steam is supplied from the boiler when the regulating or throttle valve is opened. If the valve be moved from left to right, at first the port B is placed in connection with the exhaust port C, through the exhaust cavity D in the slide-valve, so that the steam from that end of the cylinder may pass to the condenser; and shortly afterwards the steam in the casing is allowed to pass to the opposite end of the cylinder, through the port A. By the movement of the valve successively from left to right, and *vice versâ*, the steam is alternately admitted to, and exhausted from, the opposite ends of the cylinder, and the reciprocating motion of the piston is produced.

Eccentric and Eccentric rod.—The motion of the slide-valve is generally produced by means of an eccentric and rod, sketches of which are given in Fig. 74. A circular cast-iron sheave, E, has bored in it, eccentrically with its own circumference, a hole of the same diameter as the crank-shaft. This eccentric sheave is keyed firmly on the shaft, so as to revolve with it. Eccentric sheaves for marine engines have frequently to be made in halves firmly secured together, in consequence of the couplings, &c., which prevent the sheave being put on the shaft in one piece. On the circumference of the eccentric there works a ring, S, called the eccentric strap, to which the eccentric rod, R, is attached. The end, B, of the eccentric rod is connected by a joint to one end of the link which works the slide-valve rod.

In the figure the eccentric rod and one half of the eccentric strap are shown cast in one piece. Frequently, however, the rod is of wrought iron or steel, with a T end, by means of which it is secured to the gun-metal eccentric strap by studs and nuts, as shown in Figs. 69 and 94. In some examples the eccentric strap, as well

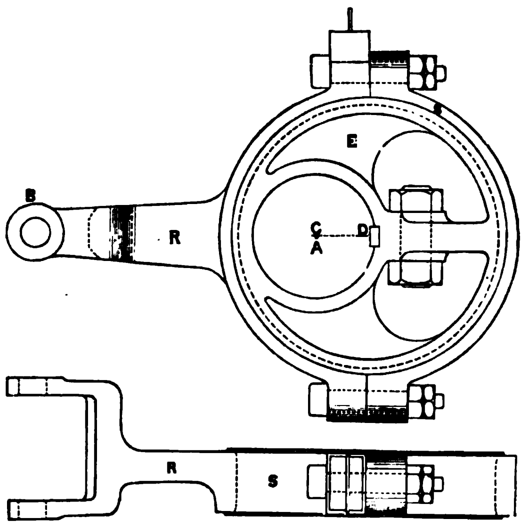


Fig. 74.

as the rod, is made of wrought iron, one half of the strap being forged solid with the rod. In these cases the straps are fitted with gun-metal liners to form the rubbing surfaces.

It is clear, that as the shaft revolves, the action of the eccentric will produce a reciprocating motion in the end, B, of the eccentric rod, and consequently in the slide-valve itself, to which it is connected by means of the link, the motion being in all respects identical with that produced by a crank, whose throw is equal to C D, the distance between the centres of the crank-shaft and eccentric

sheave. The throw of the eccentric is, however, generally so small that it would be impossible, practically, to make a crank to do the work when the valves are worked from the main crank-shaft, as this would involve the cutting of the shaft in two. The eccentric has, therefore, been adopted as the means of producing the required motion, and it will be seen that in this manner any amount of movement, however small, may be obtained.

In some cases the slide-valves have been placed on the tops of the cylinders and worked by means of a small

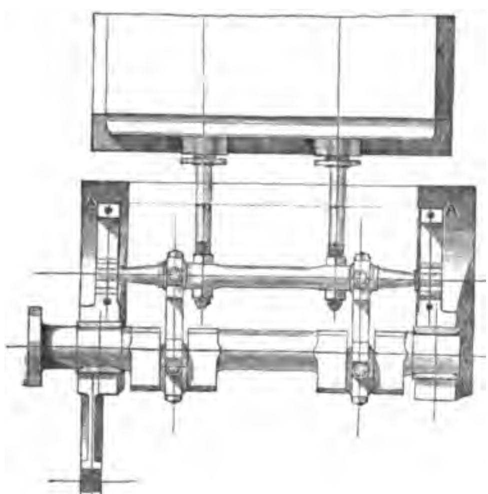


FIG. 75.

crank-shaft, driven by the engine crank-shaft, through the interposition of cogged wheels. Sketches of this arrangement, as fitted by Messrs. Maudslay in several ships of the Royal Navy, are shown in Figs. 75 and 76.

In this arrangement, spur wheels are fitted on the main and slide-valve crank-shafts, and are connected by two additional wheels, carried by a bar working in suitable guides at its two ends, to keep the wheels con-

stantly in gear with each other, as the bar is raised or lowered. By altering the position of this bar, the slide-valve crank-shaft is turned, and consequently the position of the slide-valve, relatively to the crank, is altered. All the expansion in these engines is done by the slide-valve,

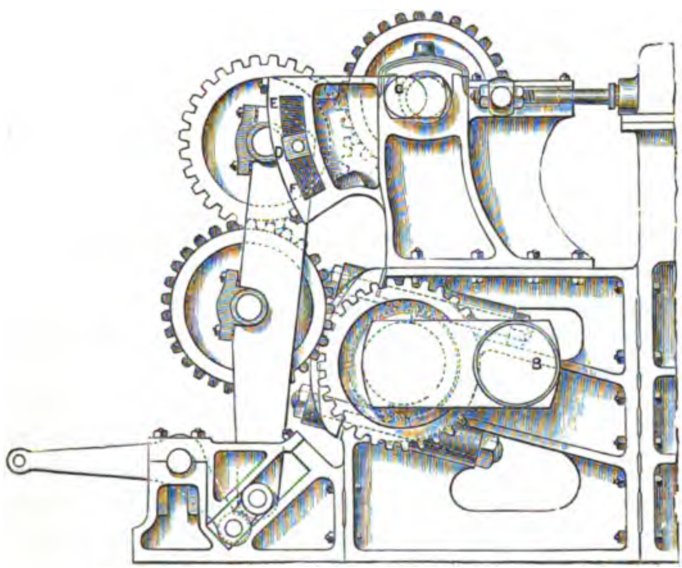


Fig. 76.

no special expansion valve being fitted, the slide-valve having three ports for admission of steam, as shown in Fig. 77, to facilitate early and sharp cut-off. For working full speed ahead, the bar is raised so as to bring the block D to the position E at the top of the slot, and for full speed astern it is lowered to the position marked F. When the block is at intermediate positions in the slot, the engines are worked expansively. For setting the slide-valve crank-shaft, when the wheels are out of gear, the engine crank should be placed on one dead centre, and the slide crank on the opposite centre, as

shown at B and C, Fig. 76. The position of the spur wheel frame should then be adjusted so as to bring the upper guide block to the centre of the curved slot, as

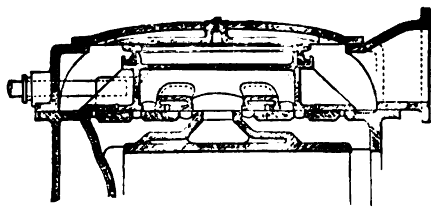


FIG. 77.

shown at D, when the wheels may be put in gear. The teeth of the wheels are numbered, so that if taken out of gear at any time, they may be readily replaced in a correct position, by taking care that the teeth marked with the same numbers work together.

Slide-valve motion.—The motion of a slide-valve is approximately an exact harmonic motion; that is, the travel or length of stroke of the valve is equal to twice the throw of the eccentric; the slide-valve is at the centre of its stroke when the eccentric arm makes an angle of 90 degrees with the line of dead points, and the distance of the slide-valve from its middle position at any given time is equal to the throw of the eccentric multiplied by the cosine of the angle that the eccentric radius makes with the line of dead points. The obliquity of the eccentric rod modifies this to some extent, especially when the rods are very short, but, in well-designed gears, the ratio of the length of the rod to that of the eccentric arm is so great that the disturbance in the motion is small, and the alteration it produces in the position of the valve is either when the ports are closed or when they are wide open to steam, at which points a slight displacement of the valve influences least the distribution of the steam.

Harmonic motion may be illustrated geometrically as follows:

Suppose the point P, Fig. 78, to move uniformly round the circle, A P B, and the point Q to move along the diameter A B in such a manner as to be at every instant vertically above or below P. Then the motion of Q would be exact harmonic motion. At the beginning of the stroke the motion of Q would be slow, gradually increasing till it reached a maximum when passing the centre O; after this it would gradually diminish till it became zero at B, when the direction of motion would change, and the point Q would travel back from B to A in a similar manner. Its distance from the centre of motion, O, at any instant is evidently equal to O P cos. P O A, which, in the case of an eccentric, would be the length of the eccentric arm, multiplied by the angle it makes with the line of dead points.

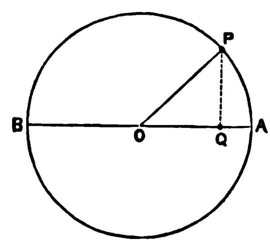


FIG. 78.

This motion is exactly realised by any point in a slot-headed sliding rod driven by a uniformly rotating crank,

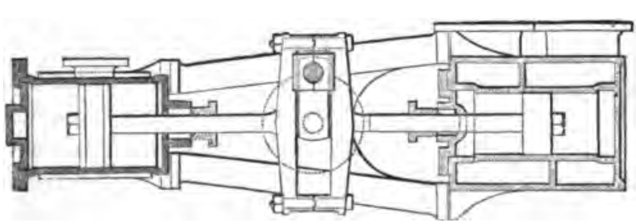


FIG. 79.

as shown in Fig. 79, which arrangement is frequently utilised in small donkey engines.

This is simply equivalent in an ordinary engine to neglecting the length of connecting rod, or, as it is more

generally expressed mathematically, by considering the length of the connecting rod to be infinite.

The same motion may be produced by means of an eccentric circle, as shown in Fig. 80.

Suppose a circular plate or cam to revolve round a centre, C, eccentric with its own circumference, and act against a flat bar, A B, at the end of the sliding rod S, which is kept moving in a straight line by suitable guides. The point of contact of the cam with the guide bar will always be vertically above the centre P, and the motion of

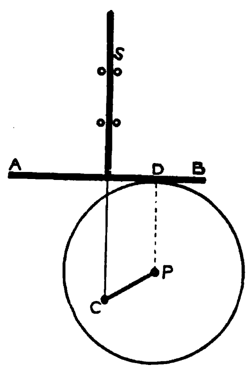


FIG. 80.

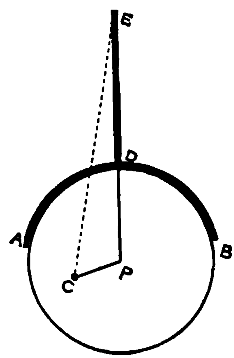


FIG. 81.

the rod will clearly be equivalent to that produced by a crank of length, C P, with an infinite connecting rod.

The step from this motion to that of the eccentric, as worked practically, is very simple, as we have only to conceive the flat bar bent into a half-hoop to fit over the circumference of the cam when, in addition to the vertical movement, the rod will also receive a swaying motion from the action of the eccentric circle. The length of the eccentric rod, instead of being infinite, is now represented by P E, Fig. 81, the end E of the rod being constrained by proper guides to move in a vertical straight line. The eccentric and eccentric-rod may, therefore, in all respects,

be regarded as a mechanical equivalent for the crank and connecting rod, and it is adopted in all those cases in which, from the smallness of the travel, it is inexpedient or impossible to obtain the motion by the direct intervention of a crank.

The most simple form of the *slide-valve* is a single-ported valve, without either lap or lead, as shown in Fig. 82. For distinction we will call this the normal valve. It is clear that in any case the slide-valve must be long enough to cover both the steam ports at the same time, or otherwise the steam would pass to both sides of the piston at once, and no motion would ensue. In the

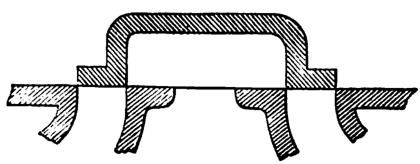


FIG. 82.

present example the valve is just of sufficient length to exactly cover both the ports.

In the figure the valve is shown in the centre of its stroke, just closing both steam ports. At this instant the crank is on its dead point, and the piston is at the commencement of the stroke, the eccentric arm being set at an angle of 90 degrees with the crank. As the crank revolves, the valve begins to admit steam to one side of the piston, and to place the other side in connection with the condenser through the exhaust passages, so that the steam behind the piston may escape. The amount of opening continues to increase, till the piston arrives at half-stroke, when the steam port is wide open. After this it begins to close, but does not quite shut till the piston arrives at the end of its stroke.

With this arrangement, therefore, the valve begins to

open the ports, both to the cylinder and condenser, at the beginning of the stroke; the ports continue open to a greater or less extent during the whole period of the stroke, and there is no expansion.

Lead.—It is usual, in order to facilitate the working of the engines, to arrange the slide-valves so that they may open both to steam and exhaust shortly before the end of the stroke. This is done by advancing the position of the eccentric arm with respect to the crank, so that all the motions of the valve may be earlier.

If this be done with the normal valve it will be seen that, although the valve opens just before the end of the stroke, yet it does not produce expansive working of the steam, as the ports are still open for the whole duration of a stroke.

Definition.—The amount of the opening of the port to steam at the beginning of the stroke of the piston is called *the lead of the valve.*

In considering the motion of the slide-valve the student will find it a very instructive exercise to draw to scale a section of the cylinder ports as shown in the diagrams, and make a cardboard model of the section of the slide-valve, so that it may be worked over the cylinder ports as desired.

Lap.—It is found in practice, that, in order to produce smooth and economical working of the engines, it is necessary to provide means for cutting off the admission of the steam before the end of the stroke. This is accomplished by giving the valve *lap or cover*, or, in other words, by lengthening the valve so as to make it more than exactly cover the ports,

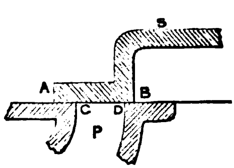


FIG. 83.

as shown in Fig. 83, in which one end only of the valve is shown. P represents the steam port of the cylinder and S the section of the slide-valve.

The side of the steam port of the cylinder at which the steam enters is called the induction or steam side, and the side at which it begins to exhaust, the eduction or exhaust side. Similar names are given to the corresponding edges of the slide-valve itself. For example, in Fig. 83, C is the induction or steam side, and D the eduction or exhaust side of the cylinder port, and A the induction and B the eduction edge of the slide-valve.

Definition.—The lap or cover of a slide-valve is the extent to which the edge of the valve overlaps the adjoining edge of the cylinder port when the valve is in the middle of its stroke. For example, A C is the lap on the steam or induction side and B D the lap on the exhaust or eduction side of the slide-valve. The lap on the steam side is often called the outside lap, and that on the exhaust side, the inside lap of the valve.

The lap on the exhaust side is always small, and is frequently absent altogether, as it is important that the communication with the condenser should be as free as possible. In some cases valves have been made with *negative lap* on the exhaust side, to increase the length of time the cylinder is open to the condenser during each stroke.

The effect of giving lap to the valve will, in the first place, necessitate an increased travel, to obtain the same amount of opening for the steam; for it is clear that the valve must travel through a distance equal to its lap before it begins to open the port at all. Secondly, in order to give the necessary lead at the beginning of the stroke, the eccentric has to be still further advanced with respect to the crank than in the case of the normal valve, and consequently all the movements of the valve will be earlier than before. The effect of the lap on the exhaust side will be to close the communication with the condenser earlier than would otherwise be the case, so that a larger

quantity of steam would be confined in the cylinder, and compressed behind the piston, until the end of the stroke.

The four principal points in the motion of the slide-valve are:

1. The instant of admission, when the steam is first admitted to the cylinder.

2. The instant of cut-off, when the steam port is just closed, so that no more steam may enter the cylinder during the stroke.

3. The instant of release or exhaust, when the communication with the condenser, the intermediate reservoir, or the atmosphere, as the case may be, is opened, to allow the steam to escape from the cylinder.

4. The instant of compression or cushioning, when the communication with the condenser, reservoir, or atmosphere, is closed, and the steam remaining in the cylinder is compressed behind the piston until the end of the stroke, so as to form a kind of elastic cushion, preventing shock and jars on the machinery, and assisting the entering steam to start the piston on its return stroke.

Angular advance of the eccentric.—In the normal slide-valve, which had neither lap nor lead, the eccentric arm was at right angles with the crank; but in the ordinary slide-valve just described, it is necessary, in order to give the valve the required lead at the beginning of the stroke, to considerably advance the eccentric beyond this position.

Definition.—The angle at which the eccentric radius stands in advance of the position that would bring the valve to its mid-stroke, when the crank is on the dead point, is called the *angular advance of the eccentric.* In ordinary engines, in which the slide-valves work parallel to the pistons, the angular advance is the excess, above a right angle, of the angle between the eccentric and crank arms.

The advance of the eccentric may be obtained approxi-

mately, as follows: Draw a circle, A B C D, Fig. 84, with the length of the eccentric arm as radius. Let O R represent the position of the crank, when on the dead centre. Set off from the centre O, O N = the outside lap, and N M equal to the lead of the slide-valve. Draw M P vertically, cutting the circumference of the circle in P. Then O P will be the required position of the eccentric radius, approximately, and the angle B O P will be the angular advance of the eccentric.

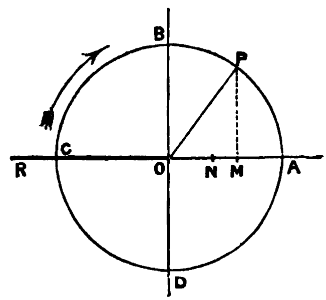


FIG. 84.

Opening of the steam port.—The opening of the port to steam, at any instant, is equal to the distance the valve has travelled from its central position, minus the lap of the valve. This is easily seen from Fig. 85. Let the dotted lines represent the valve in its central position, and the full lines its position at any given instant. Then A B represents the distance the valve has travelled:

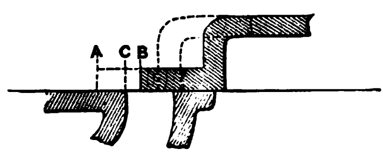


FIG. 85.

$$A\,C = \text{outside lap.}$$
$$B\,C = \text{opening of port.}$$
$$A\,B = A\,C + B\,C.$$
$$\therefore \quad B\,C = A\,B - A\,C.$$

or opening of port:

$$= \text{travel of valve} - \text{outside lap.}$$

By tracing the motion of the valve, it is easily seen that the points of admission and cut-off are regulated by

the steam edge of the valve, whilst those of release and compression are governed by the exhaust edge.

The whole motion of the slide-valve may be very clearly illustrated by means of a simple geometrical diagram, first published by Professor Zeuner, of Zurich.[1]

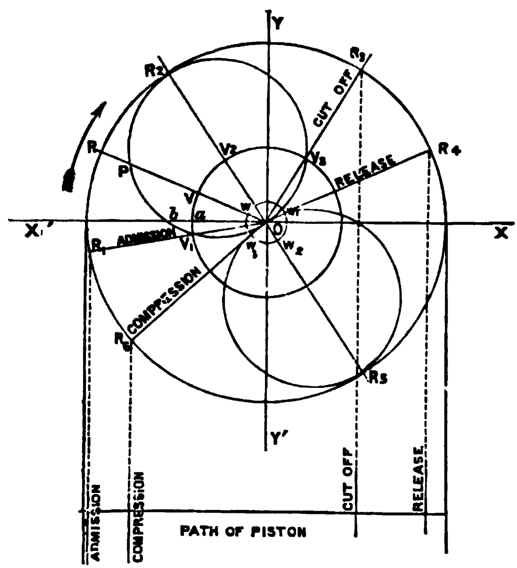


FIG. 86.

Many valuable lessons may be learnt from this diagram, which is constructed in the following manner:

Draw the lines X X′, Y Y′ Fig 86, at right angles to each other, intersecting at O. Through O draw the line $R_2 R_5$ making with Y Y′ an angle equal to the angular advance of the eccentric. The direction of motion of the crank-shaft is shown by the arrow. Make O R_2 = O R_5 = the half-travel of the slide-valve, so that $R_2 R_5$ represents

[1] An English translation by M. Muller of Prof. Zeuner's Treatise on the Motion of Slide and Expansion Valves, has been published by Messrs. E. and F. Spon.

the full travel of the valve. Describe circles on O R_2 and O R_5 as diameters; and with O as centre and O V, equal to the outside lap, as radius, describe the circle $V_1 V_2 V_3$, and with the inside lap, as radius, the circle $W_1 W_2 W_3$. Around the whole draw a circle, with O as centre, to represent the circular path of the crank-pin. This may be to any convenient scale, but the other particulars are usually, in practice, drawn full size, so that the dimensions may be measured by an ordinary two-foot rule. Then, by this diagram, the whole motion of the slide-valve can be represented. The circles on the diameters O R_2 and O R_5 are called the valve circles, $V_1 V_2 V_3$ the outside lap circle, and $W_1 W_2 W_3$ the inside lap circle.

The peculiarity of this diagram is, that for any position of the crank, say O R, the exact distance that the slide-valve has moved from its central position is given by the length O P, where the radius O R cuts the valve circle. The opening of the port to steam at this instant is given by the line P V, which is evidently equal to the distance the valve has travelled, minus the outside lap. The opening of the port to exhaust on the other side of the piston, at the same instant, is given by P W. The amounts of opening of the ports to steam and exhaust respectively, are given by the parts of the line, drawn at the assumed angle of the crank with the dead point, contained between the valve circle and the outside or inside lap circles, as the case may be.

We see, therefore, if we examine this diagram, that if a line O R_1 be drawn through the point of intersection of the outside lap and valve circles, it will represent the position of the crank when the valve begins to open the port to steam. The valve has then travelled through a distance equal to its lap and is on the point of opening. When the crank arrives at the dead point, and is in the position O X′, the travel of the valve is given by O *b*, and

a b represents the opening of the port at this point, or, in other words, the lead of the valve. The valve continues to open the port until the crank reaches the position O R_2, when the port is open to its fullest extent. It then begins to close, and when the crank arrives at the position O R_3 the port is closed. O R_3 is, therefore, the position of the crank at the point of cut-off.

We now have to deal with the exhaust side of the valve. The steam in the cylinder at the point of cut-off, expands until the crank arrives at the position O R_4, when the communication with the condenser, the reservoir, or the atmosphere, as the case may be, is opened, and the steam begins to escape, O R_4 being the position of the crank at the instant of release. The exhaust opens wider and wider till the crank reaches the position O R_5, when the valve begins to close the port, but the communication with the condenser, reservoir, or atmosphere is not entirely stopped till the crank reaches the position O R_6, which is the point of compression or cushioning. The steam remaining in the cylinder is then compressed behind the piston till the crank arrives at the position O R_1, when fresh steam enters and the same series of operations is repeated.

This simple diagram illustrates very clearly the action of the slide-valve, and if it be applied to a few typical cases many valuable deductions may be drawn. If the obliquity of the connecting-rod be neglected, the corresponding positions of the piston, at the points of admission, cut-off, release, and compression, may be found by projecting the points R_1, R_3, R_4, and R_6 on a horizontal line, representing the path of the piston, as shown in Fig. 86. The elementary knowledge of geometry which is presumed to be possessed by all students of this book will, by the application of this diagram, lead to the solution of most of the problems relating to the motion of slide-valves. By varying some of the points the alteration in

the others may easily be found, and by assuming certain elements the remainder may be determined.

As an example we would suggest that a slide-valve diagram should be drawn from the following data:

Travel of valve	8 inches.
Outside lap	2 inches.
Inside lap	¼ inch.
Angular advance of eccentric . . .	35 degrees.

Find the points of admission, cut-off, release, and compression.

Double-ported slide-valves.—When the cylinders are large it is found that the single-ported slide-valve shown in Fig. 73 would necessitate a very great travel, which would be extremely inconvenient in practice, and cause much extra work. To meet this difficulty double-ported valves have been introduced, sketches of which are given in Fig. 87. These valves are fitted to nearly all marine engines. Their action in the distribution of the steam is precisely the same as that of the single-ported valves, the only difference being in the details of their construction. The steam passage leading to each end of the cylinder, instead of terminating in a single port in the cylinder face, is divided into two parts, each being one-half the width necessary for a single port; so that the travel of the slide-valve, to admit a given quantity of steam, need only be one-half of that required for the ordinary valve.

In the single-ported slide-valve the steam is only admitted at the back, and enters the cylinder when the valve has moved a sufficient distance to allow the steam to pass from the outside of the valve to the steam port of the cylinder. The outside of the double-ported valve acts in a precisely similar manner, but in addition there is, so to speak, an inner valve, to which steam is admitted

through the passages A A, in the body of the valve itself, which it enters at the sides of the slide-valve. The inner steam ports are at the bottoms of the passages A A, and D D D D are the steam ports in the cylinder, two of which lead to each end; E is the exhaust port in the cylinder leading to the condenser, reservoir, or atmosphere, and B B B the

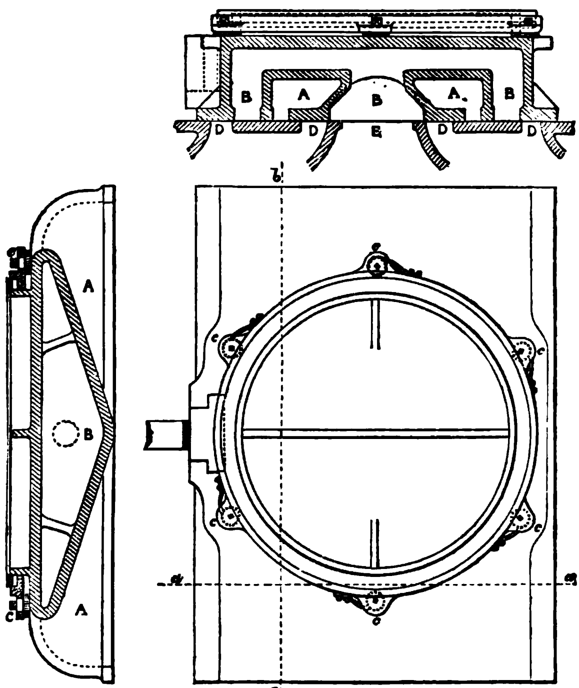


FIG. 87.

exhaust passages in the slide-valve. In the cases in which the cylinder has three steam ports at each end, as shown in Fig. 77, the travel of the valve is still further reduced for a given area of opening.

Setting slide-valves.—The fixing of the slide-valve in its proper position on the slide-rod, to insure, by its reci-

procating action, the correct distribution of the steam in the cylinder is a comparatively simple process, but one on which the efficient working of the engine will materially depend. This operation is called technically, *setting the slide-valve*, and is performed practically in the following manner.

If the motion of the valve be considered, it will be clearly seen that its half-travel, or the throw of the eccentric, is equal to the lap of the valve, added to the maximum opening of the port to steam, which represents the extreme distance that the valve moves from its central position in either direction. Slide-valves are usually arranged so as to only partially open the cylinder port to steam, say two-thirds to three-fourths, but to open it wide to exhaust.

Length of the eccentric rod.—For theoretical purposes the length of the eccentric rod is understood to mean the length measured from the centre of the eccentric strap to the centre line of the link. This must clearly be equal to the distance from the centre of the crank-shaft to the centre of the pin at the end of the slide-rod, when the valve is in the middle of its stroke.

Position of eccentric.—This is best ascertained by means of the slide-valve diagram shown in Fig. 86, from which the most suitable angle of advance can be determined with great exactness. The approximate position is, however, frequently ascertained practically by drawing on the end of the shaft, a circle with the throw of the eccentric as radius, and setting off from its centre, distances equal to the lap and required lead of the valve, as explained at p. 243. The eccentric is then secured in this position, temporarily, by means of a set screw.

Having now decided what, for the sake of convenience, we will call the fixed proportions, the actual setting of the valve is proceeded with. The whole of the rods and gear

are coupled together, the crank-shaft turned, if required, until the crank-arm is placed on its dead centre, and the slide-valve, by means of screws and nuts on the slide-rod, is fixed in the proper position to give the required *lead* for the corresponding end of the cylinder.

By the expression *dead centre* is meant, that position of the crank when it is in a line with the piston-rod, so that the pressure of the steam on the piston exerts no turning moment on the crank-shaft, but produces only a direct thrust, subjecting the shaft to bending action.

The engine is then turned round until the crank is on the opposite dead point, and the lead given by the slide-valve for that end of the cylinder measured. If the lead at the two ends of the stroke be the same, the valve will be correctly set, and all the parts may be finally fixed in position. If, however, as is frequently the case on first trial, the amounts of lead at the opposite ends of the cylinder are unequal, the position of the slide-valve on the rod must be adjusted, by means of the nuts and screws previously mentioned, until the *leads at the opposite ends of the stroke are equal to each other*, which is the condition for the correct working of the slide-valve. When this is the case the slide-valve should be securely fixed on the rod, so that its position may not alter.

If it be desired to increase or decrease the lead at *both ends* of the cylinder at the same time, this must be done by the alteration of the angular advance of the eccentric on the shaft, and not by interfering with the position of the slide-valve, which is correctly placed on the rod when the lead, whatever may be its amount, is the same at the opposite ends of the stroke.

If the centres of the slide-valve, when the crank is on each of the dead points in succession, be marked on a batten, the centre of the two points thus found will give

the central point of the stroke of the valve, which must fall exactly over the middle of the exhaust port.

In many vertical engines the lead at the lower end of the valve is made somewhat greater than that at the upper side, and more exhaust lap is allowed, to provide increased cushioning at the bottom of the stroke, for the purpose of counteracting the effect of the descending weights of piston, rods, &c. The reciprocating weights, however, are small as compared with the total steam pressure on the piston, so that the alteration from the correct theoretical working position to meet this need be very slight.

It is usual in practice to make the lead at the end nearest the eccentric rod a little more than at the other end, to allow for adjustment of the eccentric straps as they wear, which tends to draw the valve forward, and correct the original deviation from the exact position.

Packing ring at back of slide-valve.—The pressure of the steam at the back of a large flat area like that of a slide-valve would, unless special provision be made to prevent it, cause the friction between the working faces of the valve and cylinder to be intense, and bring severe strains on all the working parts of the slide gear, probably grinding the faces themselves, and quickly wearing them away. To lessen the pressure between the working faces, and thus prevent these injurious results, packing rings are fitted on the backs of the valves, as shown in Fig. 87. These rings are pressed out by the action of suitable packing or springs, so as to work steam-tight on a true surface planed on the inside of the slide-jacket cover, and thus to reduce the area on which the steam pressure can act. The hollow space in the centre of the packing ring is usually in connection with the condenser, so that, if any leakage of steam should occur, it would pass off to the condenser, and not accumulate and produce pressure on the back of the valve. In compound engines the back

of the low-pressure slide-valve only, is placed in communication with the condenser, the back of the high-pressure valve being connected with the intermediate receiver. The connection is made either by leading a pipe from the centre of the slide cover, in the space enclosed by the packing ring, to the receiver or condenser, as the case may be; or, more simply, by drilling a hole in the centre of the slide-valve itself, thus connecting its exhaust cavity with the hollow space inside the packing ring. By this fitting, the slide-valve is placed almost in equilibrium, the area of the ring being calculated so as to allow a slight excess of pressure, sufficient to keep the slide-valve steam-tight on the cylinder face.

There are many modifications in the details of fitting of these packing rings, but the principle is the same in all. In Fig. 87, the ring is shown on the back of the valve itself, and it is adjusted by means of the screws, *c c c c*, which are worked through holes in the slide-jacket cover that are closed by screwed plugs when the engine is at work. This plan is very efficient; but it is evident that the rings can only be adjusted when the engines are stopped. In many cases, therefore, the packing ring is fitted in the slide cover, the back of the slide-valve itself being simply a plane surface. This arrangement is shown in Fig. 106, and it possesses the advantage of enabling the packing rings to be adjusted at any time, whether the engines are working or not. In some instances, the ring at the back of the slide-valve is fitted to be held out against the face on the cover by the action of a single star spring, or by a thin steel disc assisted by spiral springs; but these plans have the disadvantage of being incapable of any adjustment after the cover of the slide-valve casing has been replaced.

For high steam pressures, if flat valves be used, special care must be taken to insure the efficiency of the relief

fittings at the back of the valve. Fig. 88 shows the plan usually adopted by Messrs. Humphrys, Tennant, and Co. which has proved satisfactory. A section showing the

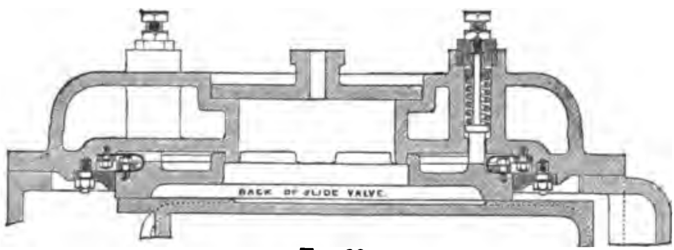


FIG. 88.

detail of the copper spring-ring which is fitted to keep the joint steam-tight is shown on an enlarged scale in Fig. 89.

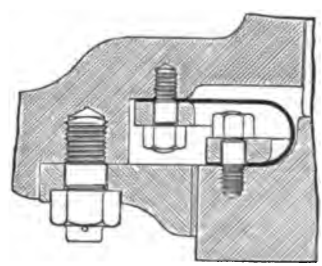


FIG. 89.

Cylindrical slide-valves.—Many engines are fitted with cylindrical instead of flat slide-valves so that the steam does not cause any pressure between the rubbing faces and no relief arrangements are necessary. Two forms of cylindrical slide-valves are shown in Fig. 90. The valve is formed by two pistons connected together, which correspond to the faces of the flat slide-valve, and work in cylindrical chambers that contain the steam-ports. The pistons are kept steam-tight by spring-rings in the usual manner, and are long enough to cover the steam-ports and allow the necessary lap. Steam is admitted from the outer sides of the pistons and exhausts into the space between them which is in connection with the eduction passages. In the form shown at A the opposite ends of the steam-chest are in communication through the tube that connects the two pistons together. In B the pipe connecting the two ends

is carried outside the cylinder. The steam-ports are stayed with bars, that run diagonally across and serve also as guides for the piston-valves.

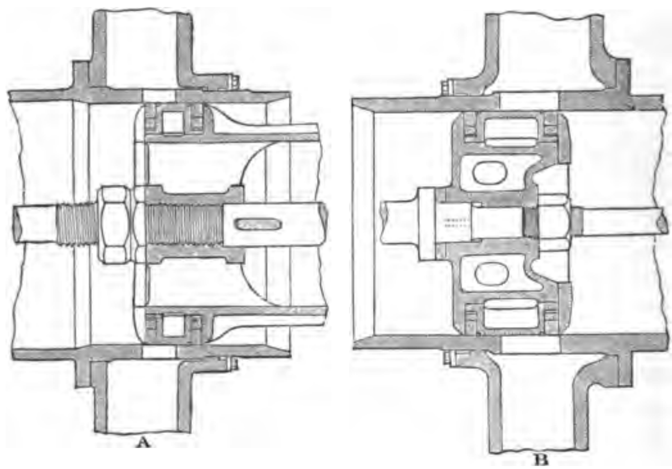


FIG. 90.

Flat slide-valves are sometimes made with two faces, front and back. This arrangement is practically equivalent to two slide-valves placed back to back. The steam is admitted to the space in the centre between the two faces of the valve and passes out through ports in the opposite faces, so that the valve is in equilibrium and no pressure is produced between the rubbing surfaces. These valves require great care in adjustment to bring the faces sufficiently close together to be steam-tight without causing them to press hard against each other.

CHAPTER XV.

STARTING AND REVERSING GEARS.

ALL marine engines must be arranged so as to be capable of being worked in opposite directions, in order that the ship may be driven either ahead or astern as may be required. It is necessary, therefore, that suitable reversing gear should be fitted to enable the slide-valves to be placed in the proper positions to produce revolution of the crank-shaft in the desired direction, whether it be ahead or astern.

Loose eccentric.—In the earlier engines, that were used in paddle-wheel steamers, this was accomplished by means of a single eccentric, which, instead of being firmly keyed to the shaft, fitted loosely on it, and was driven by stops, fixed in suitable positions on the shaft, to give either ahead or astern motion as required. A sketch of this arrangement is given in Fig. 91.

This eccentric being loose on the shaft, is balanced by means of the disc D, to prevent it falling away from its position when the slide-valve is moved by hand. The eccentric rod is attached to the slide-valve rod by means of the gab G at its end, which fits over a corresponding pin on the end of the slide-rod.

In starting these engines, the eccentric is disconnected from the slide-rod, and the slide-valves are worked by hand for a stroke or two, to start the crank-shaft revolving in the proper direction, and cause the stop for ahead or

astern motion, as the case may be, to come in contact with the eccentric, and drive it in the required direction. The gab end then drops over the pin and continues the motion in the same direction. In reversing, the eccentrics are thrown out of gear, and the slide-valves worked

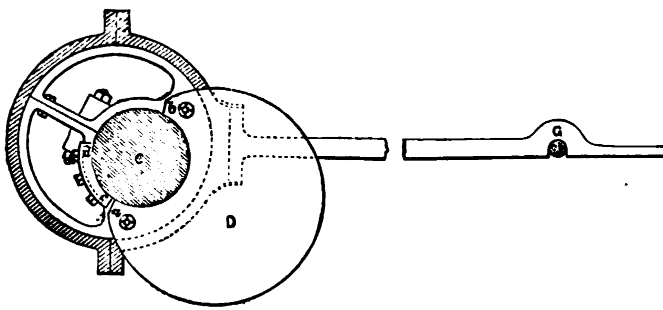


FIG. 91.

by hand to produce the reverse motion of the crank-shaft, which brings the other stop in contact with the eccentric, to drive it, and consequently, when the gab again takes hold of the pin on the slide-rod, the reverse motion of the engine will be continued. Therefore, by the aid of the stops fixed in proper positions on the shaft, one eccentric is caused to assume the correct positions relative to the crank, to produce motion either ahead or astern as may be required.

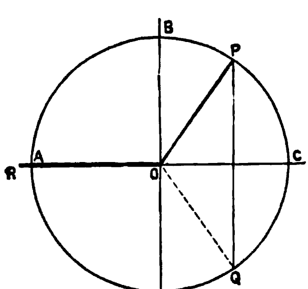


FIG. 92.

In Fig. 92, let O R represent the position of the crank on the dead point. Then, from what has been previously explained about the motion of the slide-valve, it is evident, that if the eccentric radius be in the position O P, the

crank will revolve in the direction A B C D, whilst if the eccentric arm be in the position shown by the dotted line O Q, the motion of the crank will be in the opposite direction, A D C B. The stops on the crank-shaft must, therefore, be so arranged as to bring the eccentric pulley to the position O P, or to the position O Q, according as the engine is to be driven ahead or astern.

The manner in which this is accomplished is shown in Fig. 91. On the crank-shaft, C, is fixed a stop *cd*, extending a sufficient distance round its circumference, and the ends of the balancing disc on the eccentric are arranged to form corresponding stops, *a* and *b*. When, therefore, the shaft revolves from right to left, the edge *c* comes in contact with *a*, and the eccentric and shaft revolve together so long as the motion continues in this direction; but as soon as the engine is reversed, the stops become detached, and after about half a revolution the edge *d* comes in contact with *b*, and the engine works in the reverse direction.

Link motion.—The reversing of modern marine engines is usually effected by means of the link motion, which was invented by Stephenson, and is one of the most ingenious combinations in the mechanism of the steam-engine. It is extremely simple in construction, and not only can the engines be reversed by its agency, but it provides for a considerable range of expansive working of the steam, without the interposition of any other gear.

The general construction and arrangement of this gear are shown in Fig. 93. On the crank-shaft C, there are keyed two eccentrics, one in the position to give ahead motion, and the other in the position for motion astern. The ends of the eccentric rods, which are of equal length, are attached, by working joints, to the opposite ends of a curved link L, the radius of curvature of which should be

equal to the length of the eccentric rods. This link is usually made with a slot in the centre, as shown in Fig. 93, which slides over a brass block B, to which the end of the slide-valve rod is attached.

It will be seen that when the link is lowered so as to bring the pin A in a line with the slide-valve rod, the motion of the valve will be governed by the eccentric D. If, on the other hand, the link be raised so as to bring the end of the eccentric rod B in a line with the slide rod,

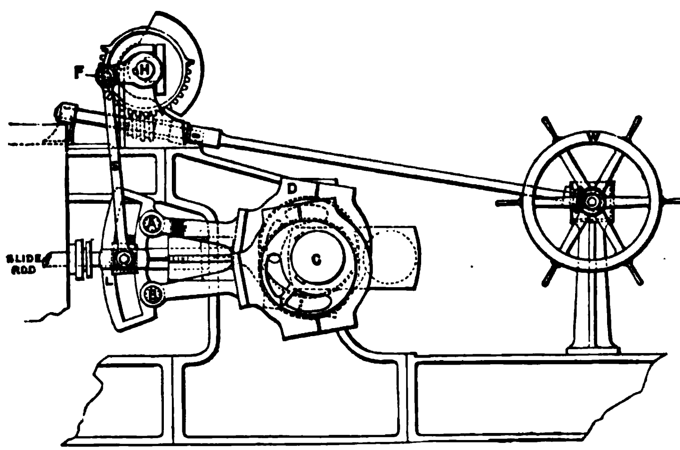


FIG. 93.

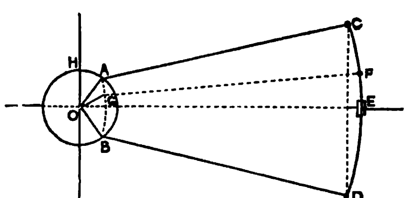
the eccentric E will govern the motion of the slide-valve, and the engine will work in the reverse direction. When the link is in the central position, though the motion of the valve would not be reduced to zero, yet the distribution of the steam would be such, that no revolution of the engine would ensue.

The centre of the link is commonly called its dead point. If the link be placed in such a position that the sliding block B is between the centre and the end of the

link, the motion of the slide-valve will be less than that when the link is at its extreme position, and the steam will be worked expansively, the motion of the engine being either ahead or astern, according to the side of the dead point at which the sliding block may be. The effect of working the slide-valve by any intermediate point of the link is equivalent to decreasing the throw and increasing the angular advance of the eccentric.

If the eccentric rods are so placed, that when the

Fig. 94.



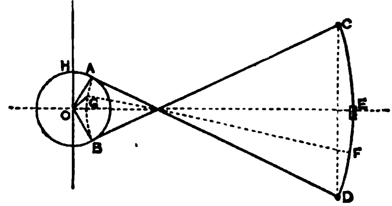


Fig. 95.

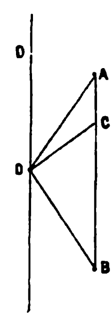


Fig. 96.

eccentrics O A and O B are inclined towards the link, the rods are in the positions shown in Fig. 94, the gear is said to have open rods. If, when the eccentrics are so placed, the rods are as shown in Fig. 95, the gear is said to have crossed rods.

The motion produced on the slide-valve when the link block E is at any intermediate position, F, of the link may be found geometrically as follows. Connect the points A and B by an arc of a circle, whose diameter bears the

same ratio to the length of the eccentric rod that the distance A B does to the distance C D. For open rods this arc should be concave, and with crossed rods convex, to the centre O. If this arc be divided at the point G, in the same ratio that the point F divides the link, the motion of the slide-valve will be the same as if it were worked, direct, by an eccentric arm O G, having an angle of advance equal to the angle H O G. O G is called the *virtual eccentric arm.*

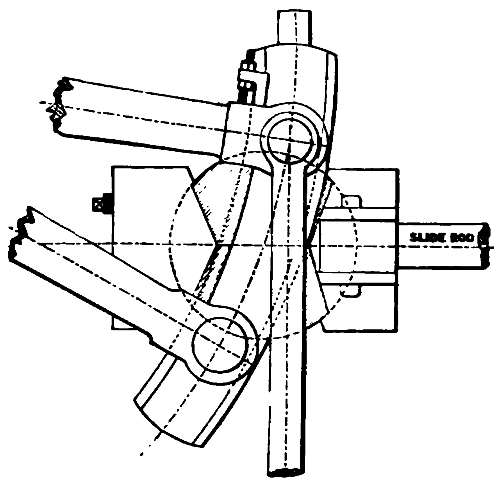


FIG. 97.

A simple approximation to the motion of the valve for any position of the block in the link, when the rods are long, may be made as follows: Let O, Fig. 96, represent the centre of the shaft, O A the ahead, and O B the astern eccentric radius. In full gear, O A is the throw of the eccentric, and the angle D O A is equal to the angular advance. For any intermediate position, divide the line A B at C in the same ratio that the block divides

the link; then the motion of the valve will be due, approximately, to an eccentric arm O C, set at an angle of advance equal to the angle D O C.

The operation of working expansively by means of the link motion is technically called 'linking up,' or 'shortening the link.' In practice, with horizontal engines, the link motion is generally arranged so that the link is lowered for going ahead, and raised for going astern, but this is not universal.

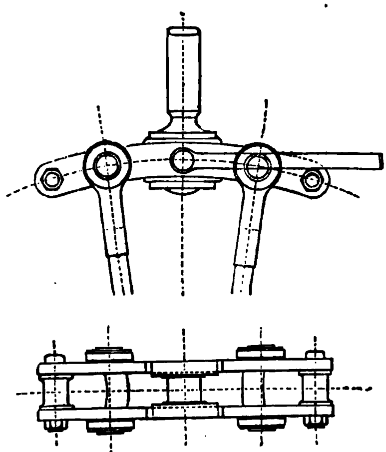


FIG. 98.

Links are sometimes made in the form of a solid curved bar, as shown in Fig. 97, instead of being slotted as shown in Fig. 93. This plan in a somewhat different form is also adopted in the vertical engine illustrated in Fig. 8. The solid link works between two segmental pieces of brass, and possesses the advantage of being capable of ready adjustment, by means of set screws, as the parts become worn by working.

In the majority of vertical engines the links are formed

of two bars, as shown in Fig. 98. In the arrangement shown the eccentric rods have forked ends and are attached to studs outside the bars, so that when the link is in full gear the centre of the eccentric rod end is in line with the slide rod. In some double bar links, the eccentric rods are attached to pins at the ends of the link between the two bars. With this arrangement the half-travel of the valve is less than the throw of the eccentric, but the details of the link itself are somewhat simpler.

These different arrangements of link motion, however, only differ in the details of construction, the principles involved being the same in all.

Starting gear.—The link is suspended generally at its centre, as shown in Fig. 93, by the suspending rod S; and during the working of the engines it oscillates about the pin F at the end of the suspending rod.

The sole object of the starting gear is to move the link into the proper position, to bring the right eccentric into action for the required motion, and this is done generally by means of the starting wheel and gear, as shown in the sketch, Fig. 93. In small engines, in which little force is required to move the slide-valves, the link is worked by means of a simple lever.

The details of starting gears vary in different engines, but the principle in all is identical, and the sketch given in Fig. 93 will be sufficient for illustration, the only variation being in mechanical details of construction. The starting wheel W, which in this case is worked by hand, is connected by worm or screw gearing to the weigh-shaft H, carrying a lever to which the suspending link S is attached; so that by the revolving of the weigh-shaft H, the link may be either raised or lowered as desired.

In powerful engines, in which the slide-valves are large, and require considerable force to move them, the starting gear is worked by steam instead of by manual

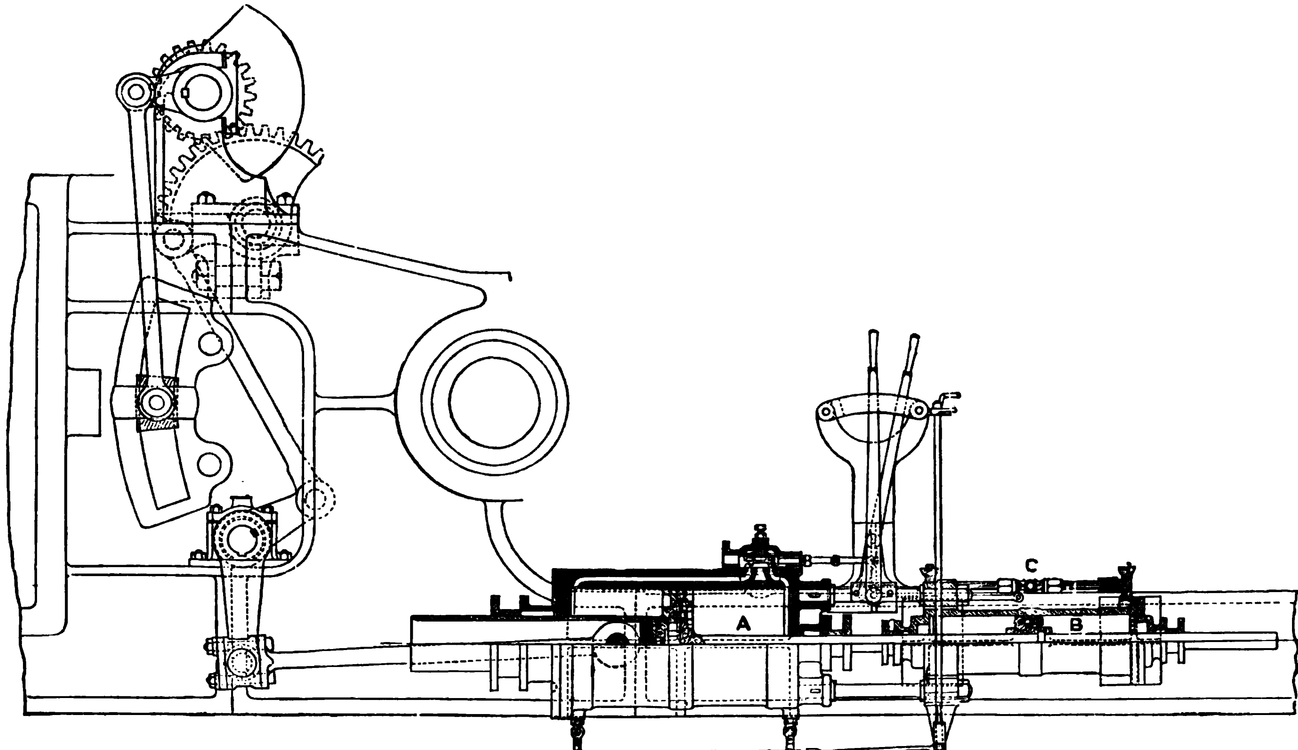


FIG. 99.

labour. This saves labour and very considerably facilitates the handling of the engines. Where steam power is available, as it always is in an engine room, it is very desirable that it should be utilised and manual labour dispensed with to as great an extent as possible. Steam-starting gear is now generally fitted to all marine engines, except very small ones in which all the operations of reversing can be readily performed by one man. This prevents the necessity of crowding the starting platform, by withdrawing men from other duties to stand by the starting gear, when going in or out of harbour, or performing steam evolutions, and renders it possible for the engines to be reversed almost instantly, in case of emergency, with only the engineer of the watch in the engine room, without waiting for the assistance necessary with ordinary hand-starting gear.

Figs. 99 and 100 show two types of steam-starting gear, which are now in general use.

In Fig. 99, the steam presses on a piston, which acts directly, through a system of levers, on the reversing weigh-shaft. In this arrangement there is, in connection with the steam-cylinder A, an oil or water cylinder, B, the two ends of which are connected by a pipe with a regulating cock, C, in it. This cylinder is filled either with water or oil, and acts as a brake on the starting gear; the velocity of the piston of the steam cylinder being governed by the amount of opening of the cock C, which regulates the rate of flow of oil or water from one end of the cylinder B to the other. This is the simplest form of steam-starting gear, and if properly fitted and intelligently worked gives satisfactory results.

In many cases, however, it is found desirable to fit a small auxiliary engine for working the link, as shown in Fig. 100. In this case all the intermediate gear is the same as in the hand-starting gear shown in Fig. 93;

the only difference being, that the small starting engine does the work of turning the starting wheel, instead of its being performed by manual labour.

In fact, we may say that whilst the steam-starting gear shown in Fig. 99, replaces the hand lever used in the

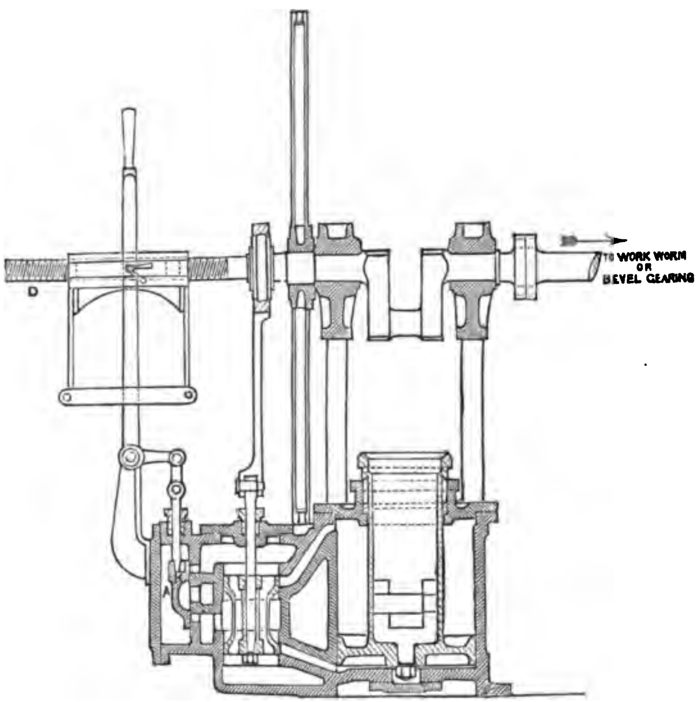


FIG. 100.

smaller class of engines, which only requires direct action, the steam-engine shown in Fig. 100 takes the place of the hand wheel to work the link by means of worm and screw gear, which require rotatory motion.

These starting engines are often fitted with automatic stopping gear, D, to prevent the possibility of the engine

over-running and straining or breaking the gear. If, however, the reversing arm on the weigh-shaft be arranged so that it is capable of continuous rotatory motion, as is now frequently the case, the automatic gear may be dispensed with. This kind of reversing engine is sometimes adapted for use in turning the main engines, and in some cases also for driving a winch barrel for lifting weights, when the engines are under examination or repair.

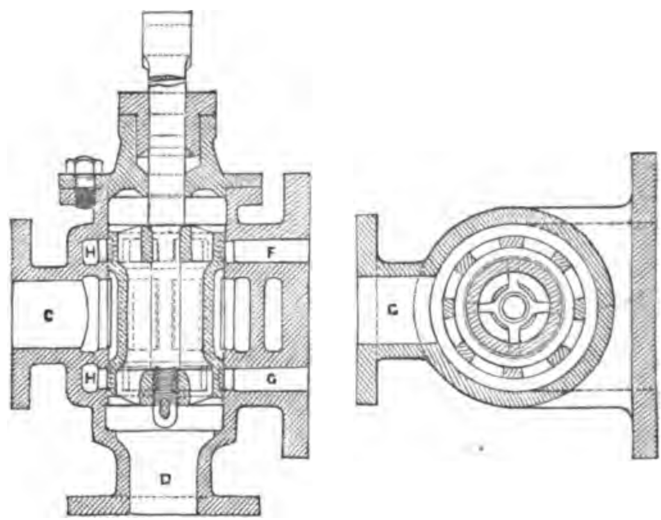


FIG. 101.

The steam for working the starting engine should be taken from a branch on the main steam pipe and not from the auxiliary steam service, the branch being on the boiler side of the regulating valve for the main engines.

Steam-starting engines themselves are generally made reversible, so as to be capable of either raising or lowering the link as may be desired. This is sometimes effected by fitting them with double eccentrics and link motion, but more generally, as shown in Fig. 100, by fitting them with

a special reversing valve, sometimes called a differential valve, sketches of which are shown in Fig. 101. If the reversing weigh-shaft be fitted so as to be capable of continuous circular motion, the steam starting engine need not be fitted with reversing gear.

Reversing gear with single eccentric.—The engine in this case is fitted with a single eccentric; and as this must be capable of working the starting engine in either direction, it must be keyed on the shaft at right angles to the crank; that is, the eccentric has no angular advance, and consequently the slide-valve must be without either lap or lead.

The reversing valve, A, Fig. 100, is worked by hand, and may be either an ordinary slide-valve, as there shown, or more generally, as shown in Fig. 101, a cylindrical valve, the action of which, however, is precisely similar to that of a flat slide-valve. In Fig. 101, C is the steam orifice leading to an annular space around the body of the valve, from which the steam may be admitted to either of the passages F or G through a series of ports opening into the belts H in the casing. The centre of the valve is open to the orifice D, which leads to the exhaust or waste steam pipe. It may therefore be seen that by raising or lowering the reversing valve, either of the ports F or G, as desired, may be placed in communication with the steam or with the exhaust pipe. The passages F and G lead to the slide-valve of the starting engine, F being in connection with the outside of the slide-valve and G with the inside, or what in an ordinary case would be the exhaust side of the valve, through the passage L, Fig. 102.

The casing of the reversing valve, therefore, has four orifices, one of which is always in connection with the steam and another with the exhaust pipe. The other two, leading to the cylinder, one to the outside and the other to the inside of the slide-valve, are made either into steam or

exhaust passages as required for the desired direction of motion of the engine.

Suppose Fig. 102 to represent the position of the slide-valve of the starting engine at the time when steam is admitted to the cylinder by the action of the reversing valve. This diagram is drawn to one-half the scale of Fig. 101. The casing outside the valve is in connection with the port F of the reversing valve casing, and the inside of the valve with the port G. It will be seen, therefore, that if the reversing valve be raised, F will be in connection with the steam and G with the exhaust passage. The steam will then pass to the casing of the slide-valve and enter the cylinder through the port K, whilst M will be in connection with the exhaust through the passage L leading to the orifice G in the reversing valve casing. The piston will therefore ascend. If, on the other hand, the reversing valve be lowered, G will be the steam and F the exhaust port. The steam and exhaust passages will consequently be changed, the steam from the port G entering the cylinder through the passage L and port M, the other port being in connection with the exhaust, so that the piston will descend and the motion of the engine will be in the reverse direction.

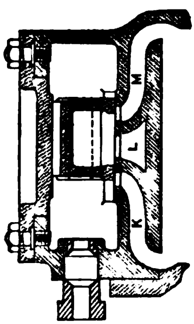


FIG. 102.

Provision must be made to prevent these slide-valves from being forced off the cylinder face, when the steam pressure is acting inside the valve and the outside is connected with the exhaust pipe through the passage F in the reversing valve casing. Owing to the absence of lap on the slide-valve this arrangement is not economical in the distribution of steam, but it is very convenient, and is largely adopted in small engines, such as starting,

steering, turret, capstan engines, &c., in which economy is not of the first importance.

Many different arrangements of gear for working the slide-valves have been designed to supersede the link motion. Several of them produce a very perfect distribution of steam at all points of cut-off, and have been extensively used in recent engines. In most of these slide-valve gears the motion of the valve is obtained by compounding two motions, one in the direction of motion of the piston and the other at right angles to it. The slide-valve rod is connected with a movable centre which traverses an arc passing through the centre of the reversing shaft, and coincides with it when in the dead position, thus securing a uniform lead at all points of cut-off. By suitably proportioning the various parts and adjusting their positions relative to each other, any degree of expansion may be obtained, or the direction of motion of the engines reversed as may be desired.

Marshall's valve gear.—This arrangement of slide-valve gear, which has been fitted by Messrs R. and W. Hawthorn to a large number of marine engines, including those of several recent ships of the Royal Navy, is illustrated in Fig. 103. In this system only one eccentric is used, the end of the eccentric rod being attached to a rod hung from a pin on the reversing shaft lever, R, by which it is constrained to move in an arc of a circle inclined to the centre line. To an intermediate point, P, in the eccentric rod a connecting link is attached which communicates the necessary motion to the slide-valve rod. By adjusting the position of the reversing lever R, any desired degree of expansion can be obtained or the engines reversed as required. In this system there are few working parts, and the distribution of steam both for full power and for expansive working is satisfactory.

Kirk's valve gear.—In this arrangement, patented by

Mr. A. C. Kirk of Glasgow, eccentrics are dispensed with and the necessary slide-valve motion is obtained by compounding two motions at right angles to each other, one derived from the piston rod and the other from the connecting rod. A skeleton diagram showing the arrangement of rods and levers to produce this motion in a vertical engine

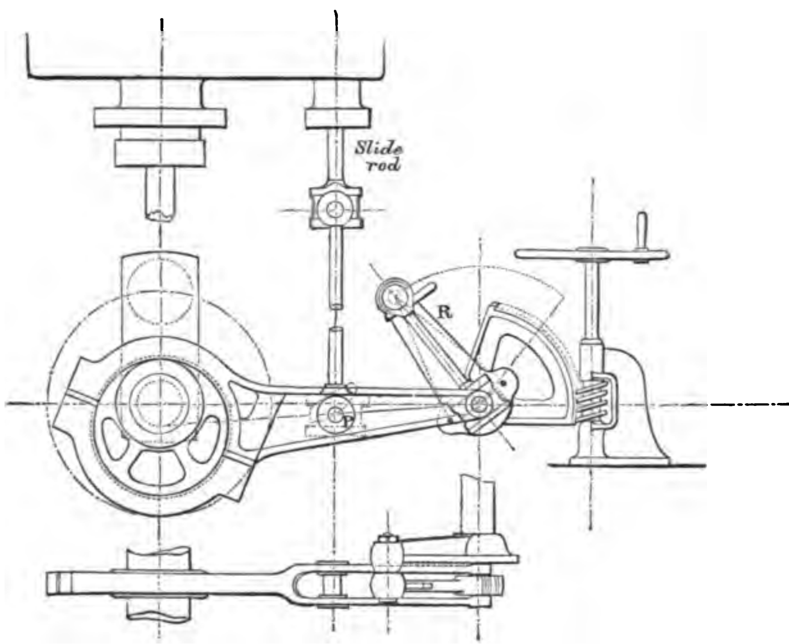


FIG. 103.

is shown in Fig. 104. A is the centre of the crank-shaft and B C is the connecting rod. The valve link or sector M N O is carried at its centre, N, by a pin on the air-pump lever T R S, from which it receives a movement in the direction of motion of the piston equal to the sum of the lap and lead of the valve. The movement necessary to give the valve its

opening is obtained by causing the link to rock on its centre N by a rod actuated by the swaying of the connecting rod at right angles to the piston rod. At the point E in the connecting rod an oscillating bent lever D E F is attached, the end F being connected to the arm L N of the valve sector by the rod F L. The other end D of this bent

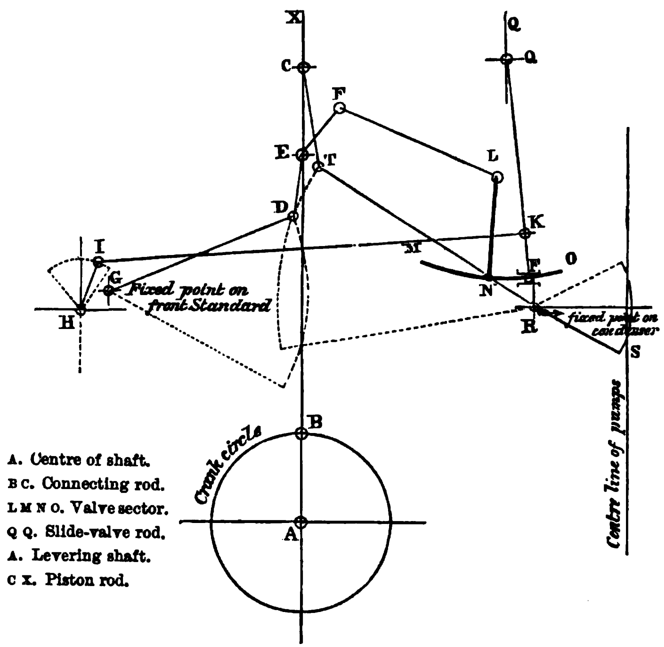


FIG. 104.

lever is connected to a fixed point G on the front standard of the engine to produce an approximately parallel motion. H is the reversing shaft and H I the reversing arm. This is connected to the radius rod Q P of the valve gear by the rod I K. Q Q is the slide-valve rod. This system of valve gear has been fitted to a considerable number of marine engines, and produces a very correct distribution of the steam.

Joy's valve gear.—Fig. 105 shows the arrangement of Joy's patent slide-valve gear fitted to the engines of H.M.S. 'Amphion' by Messrs. Maudslay, Sons, & Field. In this gear also the eccentrics are dispensed with, and the necessary movements of the slide-valve are obtained from the connecting rod. The vibrating link B, jointed to the

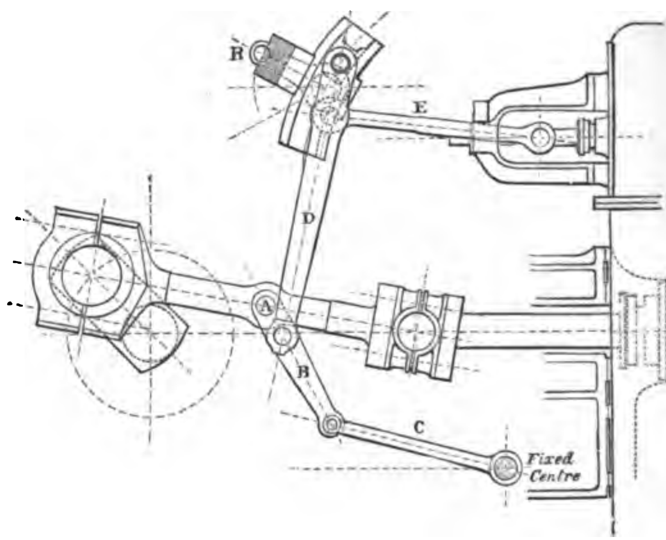


FIG. 105.

connecting rod at A, has its lower end constrained to move vertically by the action of the radius rod C. The lower end of the rod D works on a pin in the vibrating link B, and its upper end is attached to a pin in a sliding block that works in a sector carried by the reversing shaft, the centre line of the sector passing through the centre of the reversing shaft. From D the motion is communicated to the slide-valve rod by means of the link E. The forward or backward movement of the engine is governed by inclining the sector on one or the other side of the vertical

centre-line, and the amount of expansion depends on the amount of the inclination, the exactly central or vertical position being 'mid gear.' The reversing arm R is connected to the starting engine in the usual manner. The indicator diagrams taken on the official trials of the 'Amphion' were quite satisfactory at all points of cut-off. This description of slide-valve gear has been fitted to a large number of engines, both locomotive and marine.

CHAPTER XVI.

CYLINDERS AND THEIR FITTINGS.

We now pass on to describe the cylinder, in which the heat that has been stored up in the steam is partially transformed into mechanical work on the piston, and communicated to the working parts of the machinery.

Fig. 106 shows a section of a cylinder, with piston, slide-valve, &c., complete, and from this sketch the

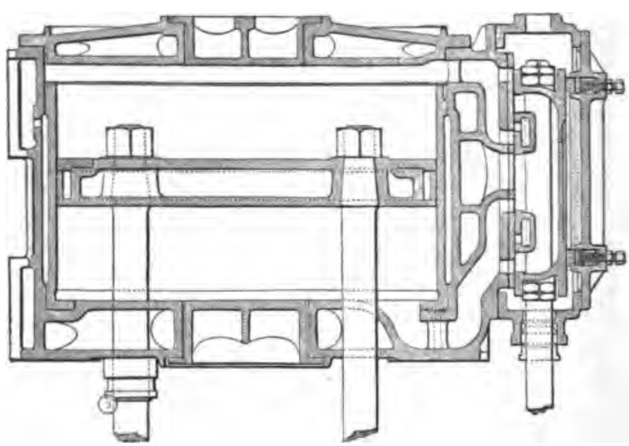


Fig. 106.

general form and arrangement may be clearly understood. The cylinder is made of cast iron, and is bored out internally, so as to form a true cylindrical surface on which

the piston works steam-tight. The inner or crank end of the cylinder is usually cast with it, and forms a part of the cylinder itself. In the centre of this end there is a large hole, which allows the boring bar to be passed through for the purpose of boring out the internal surface, and is afterwards utilised for the periodical examinations of the cylinder. When the engine is at work this hole is closed by a door or plug, which is jointed so as to be steam-tight. The open end of the cylinder is fitted with a cover, which forms a steam-tight joint on the cylinder. When the cylinder is large, the cover is fitted with a door, similar to the one on the inner end of the cylinder, to enable the examinations to be made without breaking the large joint between the cylinder cover and the cylinder. The cylinder cover and end are usually cast hollow, the space between the two plates being kept full of steam when the engine is at work, and forming part of the steam-jacket.

Cylinder-liner.—In most modern engines the working barrel is cast separate from the body of the cylinder, to which it is secured by a flange at one end. The other end, adjacent to the cylinder cover, is left free, to allow for expansion, but is packed with elastic core, asbestos, or other packing, to make the joint steam-tight. Messrs. Humphrys, Tennant, & Co. fit to the end of the high-pressure cylinder liner a copper ring of the section shown in Fig. 107, which keeps the joint perfectly steam-tight and allows the necessary expansion. The space between the liner and the cylinder is usually from one to one and a half inches in depth, and is kept filled with

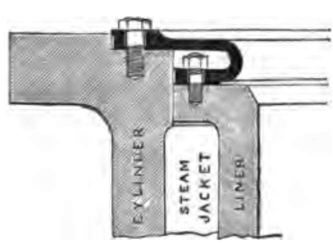


FIG. 107.

hot steam from the boiler, thus forming the most important part of the steam jacket.

There are many advantages resulting from this method of construction. It very much reduces the complexity of the casting for a jacketed cylinder. In many cases in which cylinders have been made with the inner and outer barrels in one casting, the unequal contraction of the metal in cooling has caused excessive strains on parts of the material, which developed into cracks by the working of the engines, and gave much trouble and anxiety. Not only was the efficiency of the steam jacket destroyed from this cause, but great waste of steam ensued, as there would be a constant passage of steam between the jacket and the cylinder.

With the separate liner or barrel, it is also more easy to insure that the working surface should consist of hard and sound material, so that the friction may be decreased and the durability of the cylinder increased. When the working surfaces of the cylinder become much worn, the liner may be renewed at a small cost as compared with the expense of making a new cylinder, which would be necessary if no separate working barrel were fitted. In many cases the working barrels of cylinders have been made of Whitworth's compressed steel, in which a most smooth and perfect working surface can be obtained, and the toughest and soundest material insured.

Cylinder face.—The cylinder face, on which the slide-valve works, is now generally cast separate from the cylinder, as shown in Fig. 106, to which it is secured by a number of bolts with countersunk headed gunmetal or naval brass screws, which are recessed to some depth below the working surface. These recesses act as small oil cups or reservoirs, and assist the lubrication. The advantage of this arrangement is that good sound metal can be insured for the working faces, and in case of wear the face can be

renewed without necessitating the removal of the cylinder. The faces are generally made of hard close-grained cast iron, though in some instances phosphor bronze has been used for the purpose. The composition of phosphor bronze is copper 90, tin 8·5, phosphorus 1·5 parts. This metal is hard and makes a good working surface, but it is in many respects inferior to good cast iron for cylinder faces.

Clothing of cylinders.—In addition to the steam-jacket, which is fitted to nearly all modern engines, it is necessary that the outside surfaces should be carefully covered or clothed with non-conducting material to prevent radiation. This is usually kept in place by an outer covering of wood or sheet iron, the latter material being generally preferable, as it lasts longer and can be readily taken off, without damage, to enable the non-conducting material to be renewed when necessary.

Piston.—The piston is the agent by which the energy contained in the steam is transmitted to the mechanism of the engine. Its general form, when made of cast iron, is shown in section in Fig. 106. The total depth of the piston may be taken, approximately, to be about one-tenth of the diameter of the cylinder. Holes must be left in the casting to enable the loam core, required for forming the hollow part of the piston, to be cleared out. These holes are afterwards filled with cast iron plugs tightly driven in, and secured by small screw pins.

Metallic packing ring.—It is necessary that the piston should work steam-tight in the cylinder in order to prevent leakage from one side to the other, causing waste of steam and needlessly increasing the back pressure. In most marine engines this is accomplished by means of a metallic packing ring of considerable depth, sometimes called the spring ring, which is kept pressed against the surface of the cylinder by the action of steel springs, similar in form to coach springs. Some makers use complete circular

springs of various forms to press the packing ring against the cylinder, and in other cases spiral springs are fitted instead of coach springs. This latter plan is shown in Fig. 115. In order to allow the metallic packing ring to spring and keep tight against the cylinder, it is made of a little larger diameter than the cylinder, a piece is then cut out of the circumference, obliquely, and the ring closed to fit the cylinder, so that the effort of the ring to regain its original diameter helps to keep it tight against the cylinder.

To prevent the passage of steam from one side of the packing ring to the other through the division, a plate with a tight tongue-piece is fitted behind the joint as

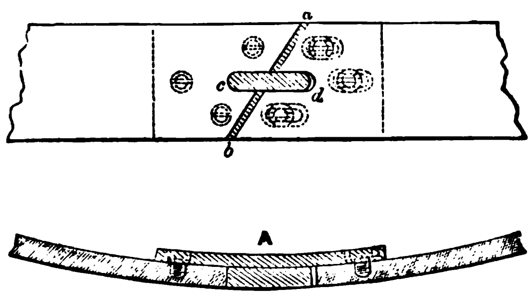


FIG. 108.

shown in Fig. 108. The ring is cut obliquely as shown at *a b*, and a groove, *c d*, is cut in the adjoining edges. A plate A, is fitted behind the joint, with a tongue-piece to fit tightly in this groove. The plate is secured firmly to one end of the metallic packing ring, whilst to the other, if secured at all, it is by means of bolts in elongated holes, so that, as the cylinder and ring wear, the ring may be allowed to expand and still keep steam-tight on the cylinder surface.

Junk-ring.—By reference to Fig. 106, it will be seen that one edge of the packing ring is in contact with a rim on the body of the piston, and that it is kept in its place

by a strong ring firmly bolted to the other side of the piston. This ring is technically called the 'junk-ring.' The edges of the packing ring, and the faces of the piston and junk-ring with which they are in contact, are scraped to a true surface, so that the joints may be steam-tight. The origin of the term junk-ring may be traced to the time before metallic packing was introduced, when the pistons were packed with hemp gasket or junk packing,

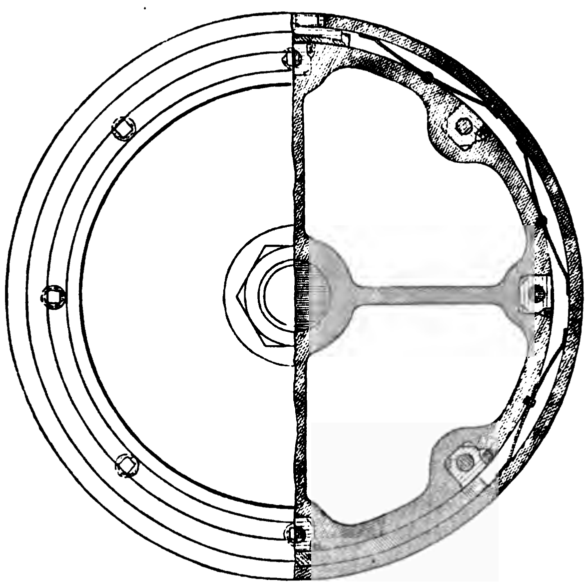


FIG. 109.

and the office of the junk-ring was to keep this packing in place, and press it against the cylinder, to keep the piston steam-tight.

Fig. 109 shows a plan of a piston with one-half the junk-ring removed, to show the arrangement of the springs, &c., and the method of attaching the junk-ring to the piston. It will be seen that brass nuts are let into the

body of the piston, into which the junk-ring bolts are screwed. This is necessary for cast-iron pistons, as the junk-rings have to be frequently taken off for the examination of the springs, and if the bolts were screwed into the body of the piston, they would soon become slack, and the defect would be difficult to remedy. With the separate brass nuts, defects are less likely to occur, and when they do they can be readily made good by the fitting of new nuts and bolts. Fig. 109 represents the arrangement for a horizontal engine. It will be seen that the springs are not continued all round the piston, but solid blocks are substituted for them at the bottom, for about one-fourth of the circumference, to assist in supporting the weight of the piston.

Guard ring.—To prevent the possibility of any of the junk-ring bolts slacking back, a guard ring is fitted to the heads of the bolts to prevent their turning, after they have been screwed up tight.

Ramsbottom rings.—In many small pistons, such as those for auxiliary engines, &c., two or more small rings, cut at one part of the circumference, are used instead of a single metallic packing ring. In this case the piston is solid, and neither junk-ring nor springs are required. Grooves are cut in the circumference of the piston into which the spring-rings fit, the joints not being placed in the same line. These are known as Ramsbottom rings. The piston of the steam cylinder in Fig. 99 is fitted in this way.

Piston rod.—The piston rod is made either of wrought iron or steel, and passes through the piston, being secured by a nut on the opposite side. The part in the piston is usually coned to allow it to be drawn up tight. It is, however, sometimes made parallel, and in this case a collar should be fitted on the rod, to take the thrust of the piston. To many steel pistons the piston rod is attached

by means of a flange. This is shown in Fig. 115. Sketches of piston rods of different kinds are given in Figs. 110 to 113.

Supporting the weight of the piston.—In large engines the pistons are of great weight, and therefore, when the cylinders are placed in a horizontal position, it is desirable

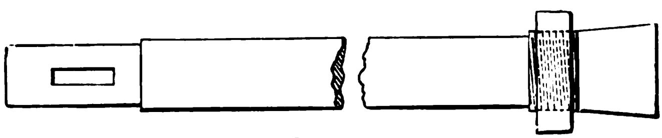


FIG. 110.

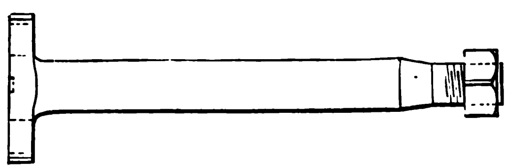


FIG. 111.

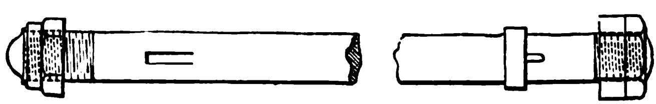


FIG. 112.

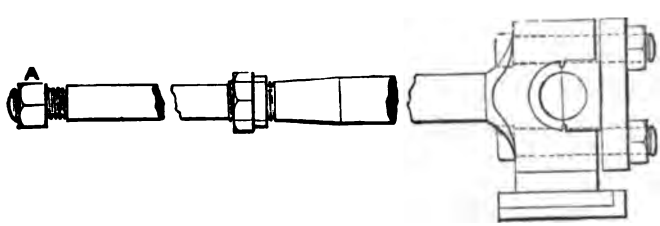


FIG. 113.

that means should be provided to support the weight of the piston and prevent the wearing of the cylinder and metallic packing ring, at the bottom, to as great an extent as possible.

In direct-acting engines with single piston rods, this is effected by continuing the rod beyond the piston,

through a stuffing box in the cover, and fitting on the end of it a brass block working on a suitable guide. The piston rod shown in Fig. 113 is of this description; the back guide block is attached to the end A of the rod, and works on a suitable guide bracket secured to the cylinder cover. By this arrangement, if the guides are properly adjusted, the whole of the weight of the piston may be carried by the piston rod, and the only pressure between the working surfaces of the piston and cylinder will be that due to the spring of the metallic packing ring.

In return connecting rod engines, which have two rods to each piston, an additional rod is sometimes fitted in the centre, at the back of the piston, and carried through the cover, with a guide block on its end working on a bracket behind the cylinder. In some cases a small supporting trunk is fitted on the back of the piston, either working through a stuffing box on the cover, as in an ordinary trunk engine, or, more generally, on a concave guide on the bottom of a hollow casing bolted on the cylinder cover, which really forms a prolongation of the cylinder. By this method the area of the piston is not reduced but, on the whole, the rod taken through the cylinder cover with a guide on the end is preferable, as it is more readily adjusted, and less liable to accident.

Piston with floating ring.—A plan designed by Mr. A. C. Kirk for relieving the cylinder of the weight of the piston is shown in Fig. 114. In this arrangement, a loose or floating ring is fitted between the metallic packing ring and the body of the piston, to keep the pressure of the springs on the packing ring. The weight of the piston is carried by the rods which pass through both ends of the cylinders. If the guides wear and the piston drops, no additional pressure is thereby produced on the cylinder, as the loose ring is not affected, and the only weight that can

be brought on the cylinder surface, is that of the metallic packing and loose rings. With the ordinary piston, if the guides be allowed to wear too much, the body of the piston will drop on the spring ring, and cause the weight of the piston to rest on the cylinder surface, which can never be the case when floating rings are fitted for sup-

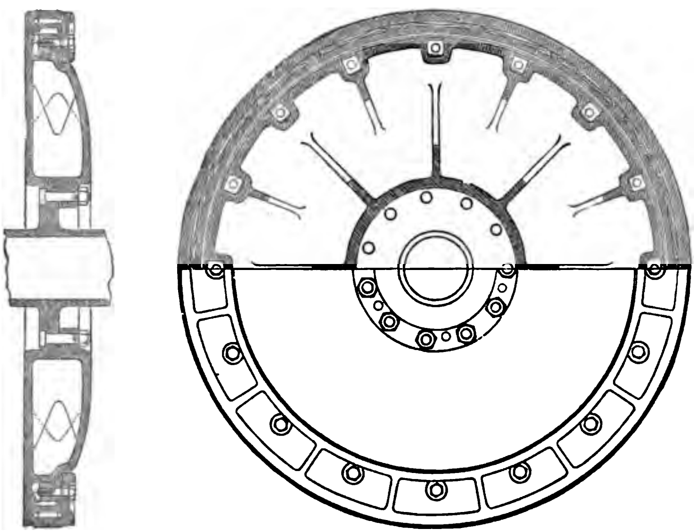


FIG. 114.

porting the springs that act on the metallic packing rings, as shown in Fig. 114.

Steel pistons.—In many engines constructed during the past few years, the pistons have been made of cast steel, and a saving of weight, of at least 40 per cent. as compared with cast iron, has been thereby effected. A sketch of a cast-steel piston is given in Fig. 115. Strong and tough steel castings of considerable weight can now be made, and as experience is gained and the processes of manufacture are further perfected, it is very probable that the use of this material will be greatly extended. It will much facilitate

the reduction of the weight of the machinery, which is of very great importance in marine engines, especially for the high steam pressures and speeds of piston and of revolution that are now employed, and will tend to promote increased safety and economy. The heaviest steel engine casting yet recorded is about 12 tons, but as demands are made, the steel makers will doubtless prove themselves equal to the occasion, as they have already done with reference to steel ship and boiler plates, &c., and there can be little doubt

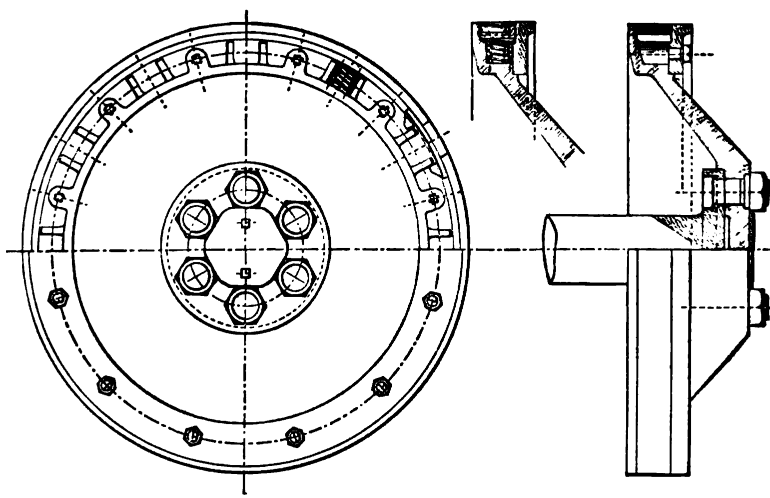


FIG. 115.

that the extended application of this material will produce an important effect on the future of marine engineering.

Stuffing-boxes.—The holes in the ends of the cylinder through which the piston-rods pass, are fitted with stuffing boxes, which, while they keep the cylinder ends steam-tight, permit the reciprocating action of the rods to take place. Stuffing-boxes are necessary in all cases in which a rod, that must be free to move, is brought through the end of any chamber that has to be kept steam- or water-

tight. This includes all rods passing through the ends of the cylinders, the slide and expansion valve rods, air, feed, and bilge pump rods, valve spindles when they are brought outside the valve-boxes, and all similar fittings.

A general arrangement of a stuffing-box is shown in Fig. 116. It consists of a cylindrical box enclosing an annular space around the rod, which space is filled with material of an elastic nature, generally known by the name of *packing*. The inner end of the stuffing-box is fitted with a brass bush, the diameter of which is a little larger than that of the rod. The gland by which the packing is pressed against the rod, so as to keep it steam-tight without unduly increasing the friction, is either of brass, or of cast iron bushed with brass, and its diameter is the same as that of the bush at the bottom of the gland.

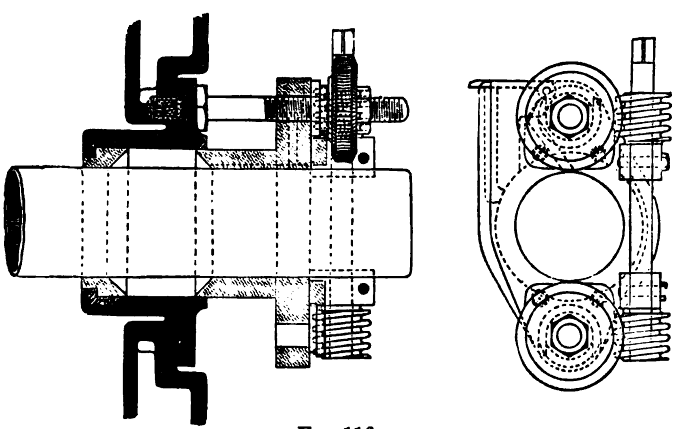


FIG. 116.

The packing used is of various kinds. Some years ago a gasket packing made by interweaving strands of hemp or cotton to form a rope, was generally used for most purposes, but for the present pressures of steam and speeds of piston, this is not suitable, though it is still used to a great extent for pump rods, cocks, &c. For piston-rods

and other similar parts elastic-core packing, which consists of a core of india-rubber round which canvas or asbestos is tightly coiled, is now generally used. For high steam pressures metallic packings of various kinds have been fitted. Beldam's packing consists of specially prepared white metal foil and canvas in alternate rings; other packings consist entirely of split white metal rings, pressed against the rods by springs, elastic core packing, or other similar means. Though apparently a minor point, the subject of the most suitable packing for the rods of steam engines is a very important one, and the introduction of a durable and satisfactory description of packing would tend to considerably increase efficiency and economy in the working of the engine.

Screwing-up gear.—The nuts for screwing up the piston-rod and other principal glands are generally fitted with toothed or worm gearing, so that they may be screwed up uniformly and adjusted with safety when the engines are at work. An arrangement of this kind is shown in Fig. 116.

Cylinder escape valves.—It is necessary that escape or safety valves should be fitted on the ends of the cylinders, so that in case of water accumulating from priming, or condensation, during the working of the engines, means of escape should be provided, to prevent excessive strains being brought on the cylinders. These valves are generally ordinary conical valves, as shown in Fig. 117, and are kept in their places by springs loaded a little above the maximum working steam pressure in the boilers. The diameter of the escape valve at each end is usually about one-twelfth of the diameter of the cylinder; and the spring should be long enough to allow the valve to open to its full extent without unduly increasing the load. In the more recent marine engines two escape valves have been fitted at each end of the high-pressure cylinders. Cylinder escape valves should be fitted with suitable guards, to prevent the danger of people being scalded by hot water from the cylinders.

Cylinder relief or drain cocks.—In addition to the escape valves, small relief or drain cocks, worked by levers from the starting platform, are fitted at each end of the cylinder, by means of which the cylinders can be cleared of water before starting the engines. These are also used for draining the cylinders when the engines are not at work, and are generally fitted to discharge either into the bilge, feed-tank, or condenser, as may be desired.

Indicator cock.—This is a three-way cock connected by pipes to either end of the cylinder, and having on the exit

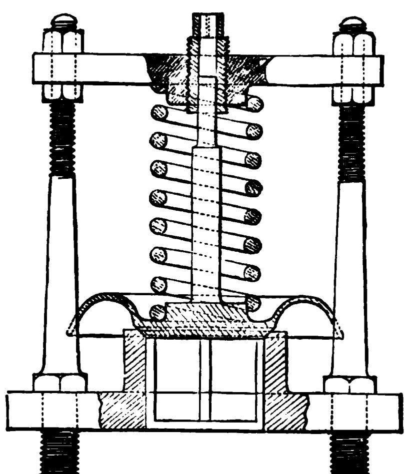


FIG. 117.

orifice a screwed socket to which the steam-engine indicator may be attached. The hole in the plug of the cock is right-angled, so that the indicator may be put into connection with each end of the cylinder in turn, and the diagrams showing the work done on the opposite sides of the piston thus taken on a single card.

Blow-through valve.—When low-pressure steam was used, it was necessary to produce a vacuum in the con-

denser before the engine could be started, as the steam pressure in the boiler was insufficient, by itself, for this purpose. To effect this, valves were fitted on the steam-pipe to enable steam to be admitted to the condenser through the exhaust pipe. These valves were called blow-through valves, the object being to blow the water, air, &c., out of the condenser through the *snifting valve,* and fill the condenser with steam. By admitting injection water the steam was condensed, and a vacuum formed behind the piston. These valves are now obsolete, as with present pressures of steam the engines can be started without the assistance of a vacuum in the condenser.

CHAPTER XVII.

CONDENSERS AND FITTINGS.

IN non-condensing engines the steam, on leaving the cylinders, passes away to the atmosphere. Most marine engines, however, are fitted with condensing arrangements, and in these the steam, on leaving the cylinders, passes through the eduction or exhaust pipes into the condensers.

Jet injection condenser.—Sketches showing the general form and arrangement of the common or jet injection condenser, with horizontal air-pump and solid plunger, are given in Figs. 118 to 121. The method of its action is explained in Chapter XII. A is the condensing chamber, into which the exhaust steam from the cylinder enters through the eduction pipe, B. The capacity of the condensing chamber in Watt's original engine was about one-eighth that of the cylinder, but in later practice it varied from one-quarter to one-half that of the cylinder, and was in some cases even larger than this. C, C, Figs. 119 and 120, are the sea-injection valves, on the side of the condenser, which are worked by levers from the starting platform. The pipes from these valves lead into one, which is in connection with the sea by means of a Kingston valve and sea-cock on the ship's side. When the engines are at work, the Kingston valve and sea-cock are kept open, and the amount of water admitted to the condenser, through the internal perforated pipes, D, is regulated by the extent of opening of the injection valves, C, C. It is necessary to

Fig. 118.

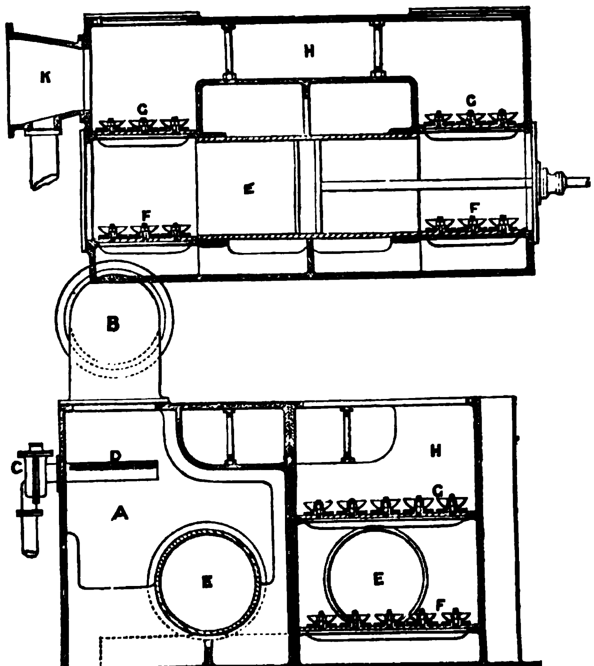
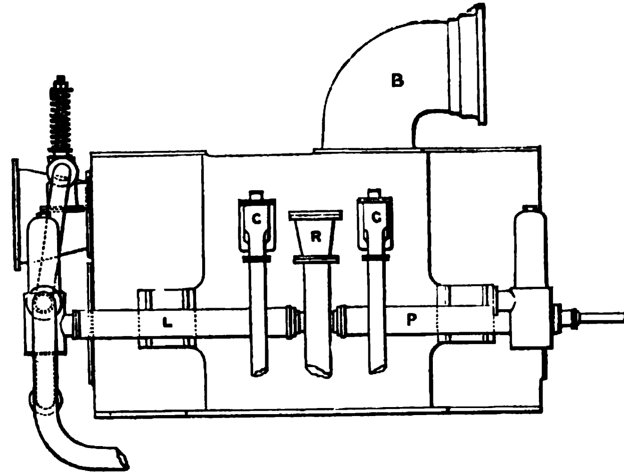


Fig. 119.

Fig. 120.



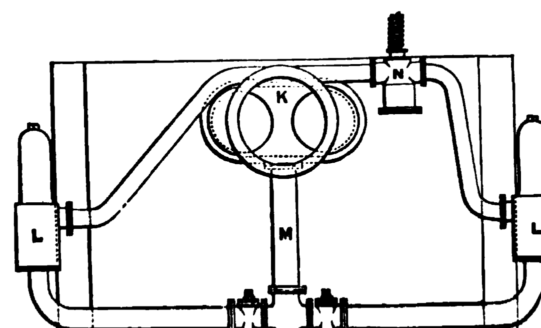


Fig. 121.

admit the water through a series of small holes in the internal injection pipe, to facilitate its mixing with the entering steam. In most condensers the sea injection is admitted through a single valve. In the example taken for illustration two valves have been used for the better distribution of the water, the condensing chamber being of considerable length.

Air-pump.—The air-pump, which in horizontal engines is generally worked direct by a rod attached to the piston or piston-rod of the engine, is marked E on the diagrams. In some cases, more especially in recent ships, the air-pump is worked by an independent auxiliary engine. By the action of the air-pump the mixed injection and condensed water, air, vapour, &c., are pumped into the hot-well, H; F, F, being the foot-valves, and G, G, the delivery or head-valves, of the air-pump. The air-pump plunger is often packed with hemp or cotton gasket, which is kept pressed against the pump barrel by the action of a suitable gland. Frequently, however, metallic packing rings are fitted.

The space between the foot and delivery valves of the air-pump should be made as small as possible to insure the whole of the air and vapour being expelled at each stroke, and to prevent accumulation of pressure in the chamber. This is especially important in the case of surface condensation, in which a much less quantity of water is acted on than with jet injection. The area through the foot and delivery valves of the air-pump varies from one-third of the area of the pump to equality, according to the speed of the engine, being the greater for the higher speeds. The velocity of the water through the passages should not exceed ten to twelve feet per second. The seatings for the foot and delivery valves are generally made of gun-metal, and are secured by bolts to suitable flanges cast on the condenser to receive them. The valves are generally made of vulcanised india-rubber,

and a brass guard, attached to the seating by a stud and nut, is fitted over them, to regulate the amount of opening, as shown in Figs. 118 and 119.

Vertical air-pump.—In vertical engines, the air-pump is generally fitted at the back of the condenser and worked by a lever actuated by the piston-rod as shown in Fig. 149. In

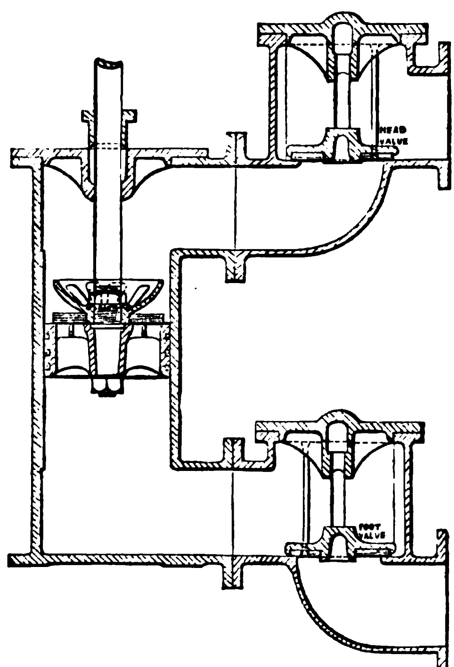


FIG. 122.

the vertical engines made by Messrs. Humphrys, Tennant, & Co. the air-pumps are worked direct from the pistons as in horizontal engines. This arrangement is shown in Fig. 8. Vertical air-pumps are usually single-acting, and fitted with bucket plungers, which have valves in them to allow the water, vapour, &c. to pass from the lower to the upper side of the plunger as it descends. This arrangement is

practically more efficient than the horizontal pump with solid plunger, as the air and vapour can be more completely exhausted from the condensing chamber. Fig. 122 is a section through a vertical air-pump, showing the details of the fittings.

Discharge pipe and valve.—A large pipe, called the discharge pipe, is bolted to the orifice K connecting the two hot-wells, and leads to a self-acting valve on the ship's side, to allow the surplus water, not required to supply the boilers, to pass overboard. This valve is called the main discharge or waste-water valve. In large engines there is generally a separate discharge pipe and valve for each condenser. In both the eduction and hot-well discharge pipes, *expansion joints* should be fitted, to allow of slight alterations in the lengths of the pipes, and prevent straining from the working of the ship or from the expansion and contraction of the parts when the engines are at work.

Feed pumps, pipes, &c.—From the hot-well the feed pumps draw their supply for the boilers. In Figs. 120 and 121 L, L, are the feed pumps, and M the feed suction pipe, which is attached to the underside of the hot-well discharge orifice K. Cocks or valves are fitted in the branch suction pipes leading to each pump to enable the supply to the pumps to be regulated. N is the feed escape valve, which is fitted on the delivery pipes from the pumps to prevent excessive pressure occurring in the pipes and pump boxes, if the feed valves on the boilers be closed suddenly when the engines are at work. The water in this case would lift the escape valve, which is kept in its place by the action of a spring, and return to the hot-well. Pressure gauges are fitted on the delivery pipes from the feed pumps.

In cases in which the feed escape pipe is carried to the suction pipe of the pump, instead of to the hot-well direct,

the *feed suction valve* should be self-acting, as shown in section in Fig. 123. This kind of valve is only kept shut by the pressure of the water in the hot-well, and it will be seen that, in the event of water being forced through the escape valve, the suction valve would be pushed open to allow the escaped water to return to the hot-well. The necessary amount of opening for feed purposes may be regulated by the screwed spindle, which is not connected to the valve but only in contact with it, so as not to impede the further opening of the valve in case of water passing the escape valve from excessive pressure.

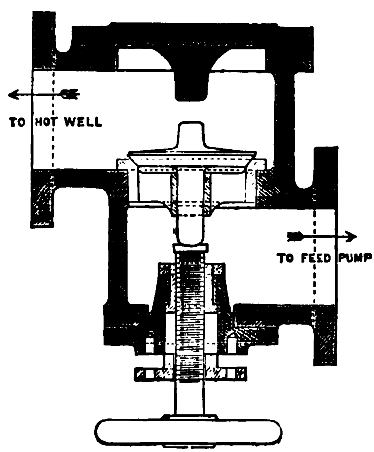


FIG. 123.

Fig. 124 shows a plan and elevation of feed pump with metal valves, the pump box and valves being shown in section. Air vessels are fitted, to prevent shock from the reciprocating action of the pump and to promote uniformity in the rate of delivery. Another arrangement with india-rubber valves is shown in Fig. 125. In this plan the engine bilge pump and valve box are in the same casting with the feed pump and valve box. India-rubber valves, however, do not appear to be generally suitable for feed pumps, especially for high working pressures of steam, and consequently, in the majority of cases, the valves of the feed pumps are made of gun-metal or lignum vitæ. The valves should be large enough to allow the necessary quantity of feed water to be pumped through with a small lift, so as to reduce the wear from the beating of the

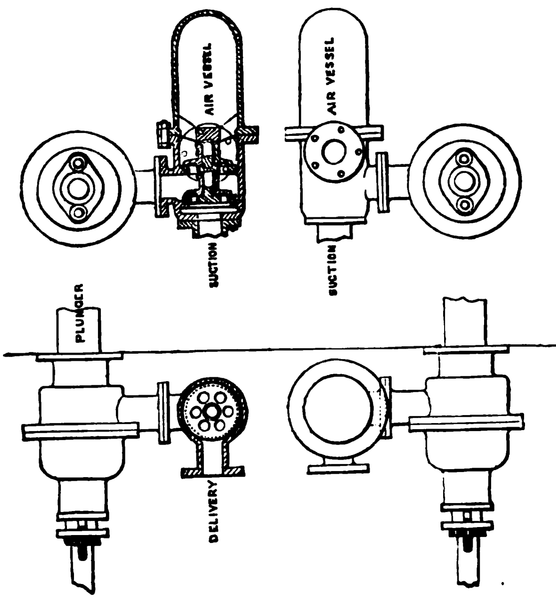


FIG. 124.

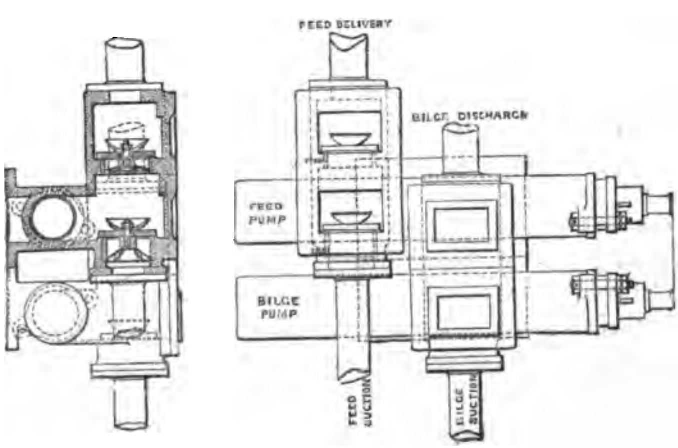


FIG. 125.

valves on the seatings when the pumps are at work. In horizontal engines the feed pumps are usually worked direct from the engine piston, either by independent rods or from a bracket on the piston rod, as may be the more convenient. In those vertical engines, in which the pumps are placed at the backs of the condensers, the feed and bilge pumps are generally fixed on the sides of the air or circulating pumps and worked by brackets on the levers for these pumps.

Bilge pumps.—The engine bilge pumps when worked from the piston are generally similar in construction and arrangement to the feed pumps as shown in Fig. 125. In the case shown in Fig. 120 one plunger is made to do duty for both feed and bilge purposes, the plunger passing through both pump barrels, one end working the feed and the other the bilge-pump valves. The bilge-pump barrel is marked P in Fig. 120. The suction pipes of the engine bilge pumps are led to different parts of the bilge in the engine department, and should be fitted with non-return valves and strainers at the ends. The delivery pipe is led to a self-acting valve on the ship's side, called the bilge-pump discharge valve.

Independent feed and bilge engines.—In the more recent ships of the Royal Navy, independent feed and bilge pumping engines have been fitted instead of attaching feed and bilge pumps to the main engines. This system has proved most satisfactory for the high speed engines and forced draught boiler rooms which are now fitted in war-ships. Feed pumps when worked by the main engines, especially by high speed engines, are somewhat spasmodic in their action, and the pressure in the feed pipes fluctuates very considerably during each stroke. With separate feed engines their speed is regulated independently of that of the main engines, and is governed solely by the requirements of the boilers, so that the pressure in the feed

pipes is kept very uniform. This enables a steady flow of feed water to be maintained which reduces the tendency to priming when the boilers are forced. By this system too the feed water is supplied to the boilers practically free from air, and this is conducive to their durability.

The feed engines, main and auxiliary, are usually fixed in the stokeholds so that the person in charge of each set of boilers has full command of the feeding of his boilers, which is an important feature in ships that are necessarily subdivided into separate watertight compartments. The main feed pumps are fitted to draw from the feed tanks only, but duplicate sets of engines and pumps are fitted, which are arranged to draw both from the feed tanks and from the sea, thus making provision for the event of break-down of any feed engine or pump. Each of the feed engines, main and auxiliary, is arranged to supply any of the boilers, and the pipes and valves from the auxiliary feed pumps are entirely independent of those from the main feed pumps.

Bilge injection.—To all jet condensers an additional injection valve is fitted, the inlet pipe to which leads to the engine-room bilge. This is called the bilge injection valve and is marked R in Fig. 120. In the case of a serious leak, and the bilges becoming flooded with water when the engines were at work, the necessary water for condensing the steam could by this means be taken from the bilge, instead of from the sea direct by the ordinary injection valve, and the air-pump thus utilised for pumping out the ship. By reference to Chapter XII. it will be seen how large a quantity of water could be thus disposed of, and this without taxing the engines, as the water would be necessary for their working. Bilge injection pipes should have non-return valves fitted in them to prevent the possibility of water passing into the bilge from the sea, and their ends should be in accessible positions and

carefully protected with strainers, to prevent chips &c. getting into the condensers and choking the valves when injecting from the bilge. In iron, steel, or composite ships, these strainers, and the lower ends of the bilge suction pipes should be made of zinced iron or steel.

Snifting valve.—The snifting valve is a non-return valve, still sometimes fitted to condensers, which is kept in its place simply by its own weight. In the early days of marine engineering, when the working pressure of steam was only a few pounds above the atmosphere, this valve was necessary to enable a vacuum to be produced for starting the engines; the air, water, &c. in the condenser being blown out through it by the steam admitted by the blow-through valve on the cylinder. It is now no longer required for this purpose, and is therefore seldom fitted, but it may be useful in jet injection condensers, especially when the discharge valves are leaky, as any water that passes into the condenser when the engines are standing will escape through the snifting valves into the bilge, instead of filling the condenser and passing over into the cylinders. The snifting valve is, however, practically an obsolete fitting.

Surface condensers.—The general arrangements of surface condensers are shown in Figs. 126 to 129.

In the condenser shown in Figs. 126 and 127, the steam entering the condenser casing from the eduction pipe, surrounds a number of small brass tubes, through which cold sea-water is drawn by the action of a reciprocating circulating pump A, worked by the piston of the main engines. Guide plates are fitted to cause the water in its passage through the condenser, from the inlet orifice to the pump, to circulate through the whole of the tubes in the direction shown by the arrows. The steam is condensed by the cold surfaces of the tubes, and the condensed water flows into the chamber B, from whence it

is pumped by the air-pump C, to the hot-well D, in the same manner as in jet injection condensers. The construction of the air-pump, valves, &c. in this condenser are precisely the same as those of the *circulating pump* which is shown in section. In all surface condensers care should be taken to insure the delivery valves of the air-pump being always kept covered with water. The orifice for the feed suction is generally kept a sufficient height above the valves for this purpose. The small sluice valve shown in the circulating pump chamber, Fig. 126, is fitted to allow water to return from the discharge to the suction side of the pump when the engines are worked at low powers, and thus to reduce the quantity of water drawn through the pump without partially closing the sea inlet.

Surface condensers have generally been fitted so that they may be worked, if necessary, as jet condensers, a sea injection valve being fitted to the exhaust steam chamber, at the entrance to the tubes. In Figs. 126 and 127 the sluice valve underneath the tubes, marked S, is fitted for the purpose of turning the circulating pump into an additional air-pump, when jet injection is used. It will be seen that by opening this valve and closing the sea suction, the circulating pump will draw from the condensing chamber instead of from the sea, and the chamber H will become an additional hot-well.

Figs. 128 and 129 show an arrangement of surface condenser, in which the water is circulated by means of a *centrifugal pump*, worked by an independent auxiliary engine. This plan is more generally adopted than the reciprocating pump worked direct from the engine piston, and possesses many advantages. The circulating pump can be kept working and the condensers kept cool when the main engines are standing; whilst in the case of the reciprocating pump, circulation ceases when the engines are stopped. In many cases with reciprocating pumps, it

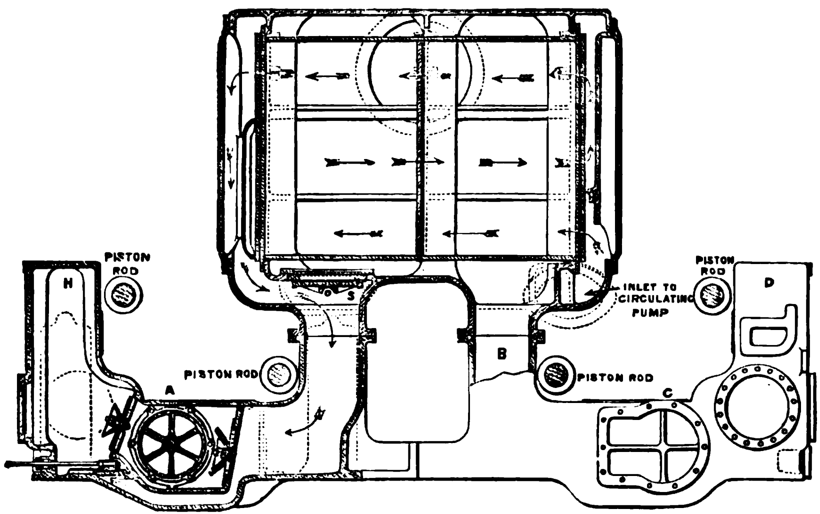


FIG. 126.

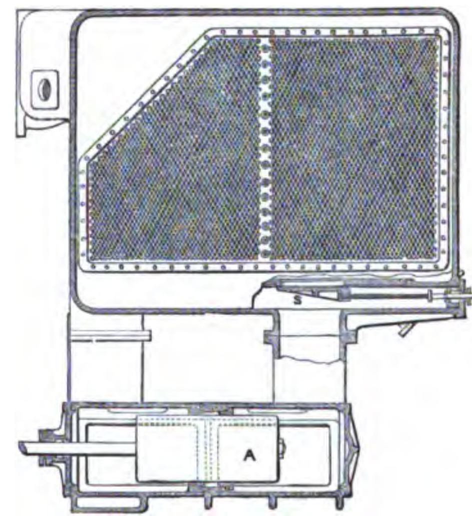


FIG. 127.

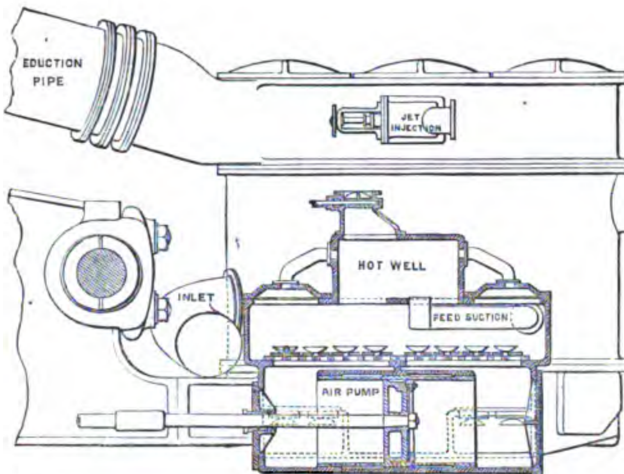


FIG. 128.

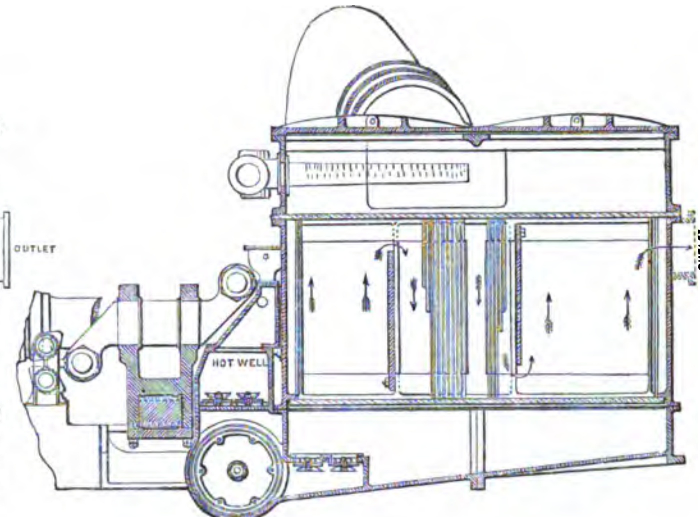


FIG. 129.

has been necessary to fit suction pipes from the condenser casings to one of the auxiliary pumping-engines in the engine-room to prevent the condensers getting hot before starting the engines. It is also difficult when the pumps are worked by the main engines to vary the amount of circulating water if required, the speed being necessarily the same as that of the main engines. Often at the highest and lowest speeds the pumps do not work so efficiently as at moderate speeds, and the vacuum is consequently decreased. These pumps are also less readily available for pumping out the ship in case of emergency, as they cannot be worked except when the ship is under way.

In surface condensers the steam is either carried through or around the tubes, at the fancy of the designer. If the steam passes inside the tubes, the water being in the space outside, the grease left on the tubes by the steam can be readily cleaned off by passing brushes through them; whilst with the system in which the steam is outside the tubes, the only method of effectually cleaning them is by filling the condenser with a solution of potash and boiling it. They may, however, be cleaned to some extent when under way by working the condensers rather hot for a while to loosen the grease, and then using the jet injection for a short time to wash it away. The plan of condensing the steam inside the tubes has answered well when the tubes were placed vertically, but with horizontal tubes it is more difficult to give the water proper direction to insure its coming in contact with all the surface. These condensers have also the disadvantage of holding a larger quantity of water than those in which the water passes through the tubes, and this weight of water has to be added to the total weight of machinery when the engines are at work. The condenser casings are in contact with the circulating water, which causes them, when made of cast iron, to corrode and decay rapidly. The system of

causing the water to pass through the tubes, the steam being in the space surrounding them, is the more general. Less weight of water is carried, and more definite direction can be given to it to insure contact with the whole of the tube surface, and though the casings are somewhat hotter in consequence of containing steam instead of water, the heat can be prevented from seriously affecting the engine-room by suitable lagging.

The area of the condensing surface in surface condensers was formerly made about the same as that of the heating surface of the tubes in the boilers, or, say, from 2 to 2½ square feet per indicated horse-power. It has, however, been found by experience that considerably less than this is sufficient, and in most modern ships the area of cooling surface is only from 1·25 to 1·5 square feet per indicated horse-power. It is very important in all surface condensers to fit circulating plates to give definite direction to the condensing water to insure its coming in contact with the whole of the tube surface.

In surface condensers the only water that accumulates in the condensing chamber is that from the condensation of the exhaust steam, and consequently the air-pump might be made very much smaller than for the ordinary jet condenser. In order, however, to provide for the contingency of working with jet injection in condensers so fitted, the air-pumps for surface condensers have usually been made nearly as large as in the case of ordinary jet condensers; the capacity, for double-acting pumps, being generally about one-twelfth the capacity of the cylinder, instead of one-ninth which is the usual proportion for jet condensers. In some cases valves and pipes have been fitted to allow water to return from the hot-well to the air-pump, if necessary, to more completely fill the chamber and assist in the expulsion of the air and vapour when working with surface condensation.

Centrifugal pumps.—Fig. 130 shows an arrangement of pump and engine used for circulating water through surface condensers. Fig. 130*a* shows a section of the pump. The water enters at the centre and passes off at the circumference by centrifugal action as the pump revolves, the direction of motion being shown by the arrow.

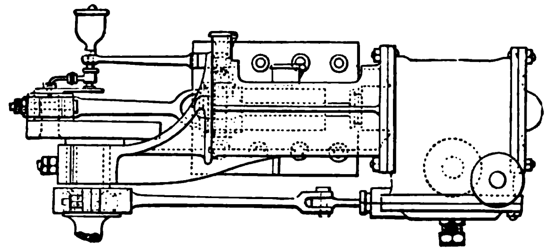
Centrifugal pumps are very useful when large quantities of water have to be pumped with a comparatively small lift, as is the case in marine engine condensers, in which both the inlet and outlet orifices are generally below the surface of the water, and consequently the only work the pump has to do is to overcome the friction of the passages and keep the water in motion. They work very smoothly, and have the advantage, that if they should be started before the outlet valve is opened there is no fear of injury to the condenser, as the pump will only churn the water and not bring great and sudden pressures on the passages. When the discharge orifice in centrifugal pumps is not at the highest part of the casing, it is necessary to fit an air-cock at the top to let off the air and prevent accumulation, as the pump must be kept quite full of water to insure its efficient action, the presence of air seriously interfering with its working.

The engine for driving the centrifugal pump as well as the steam starting engine should be worked by steam taken from a branch on the main steam pipe, and not from the auxiliary steam service.

Circulating inlet and outlet valves.—The circulating pumps take their suction from a large inlet valve on the bottom of the ship, inside which there is usually fitted a sluice valve in lieu of a sea-cock, in consequence of the size of the pipe. The discharge is through similar valves on the ship's side.

The hot-wells in surface condensers fitted with jet or bilge injection arrangements require discharge pipes and

Fig. 130.



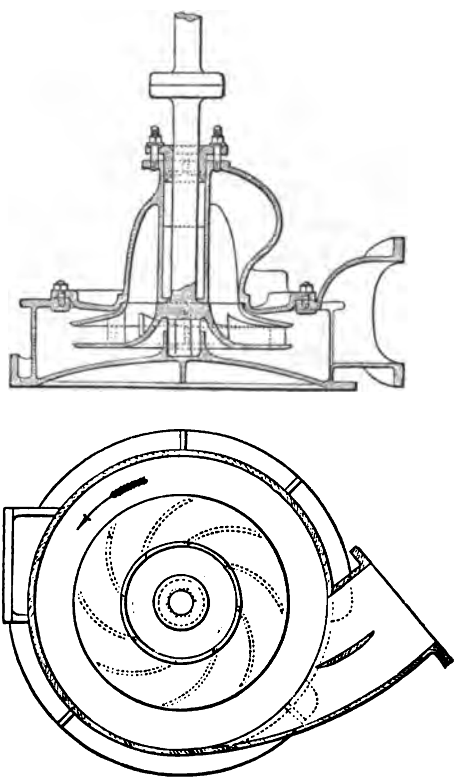


Fig. 130*a*.

valves as in the case of common jet condensers. Sometimes the discharge pipes from the hot-well and from the circulating pump lead to separate discharge valves on the ship's side. Frequently, however, non-return valves are fitted close to the discharge orifices from the circulating pump and the hot-well respectively, and these valves deliver into a common discharge pipe which conducts the water to a single self-acting discharge valve on the ship's side.

Air-cock on hot-well.—In the jet condenser, when the engines are at work, the hot-well is in connection with the sea, and therefore with the atmosphere, by means of the main discharge valve, so that air, vapour, &c., may pass freely away and not accumulate in the hot-well. In the surface condenser the case is different, as the hot-well discharge valve is kept closed, and if no relief were provided for the air and vapour, it would accumulate in pressure and prevent the air-pump from working efficiently. To provide for this, air-cocks or valves are fitted on the hot-well to allow the air to escape. These usually deliver into open funnel-mouthed pipes leading into the bilge. The delivery orifices of these air valves are kept in sight, to prevent the danger of water passing into the bilge unnoticed. If working with jet injection, when the hot-well will be connected with the sea, these cocks or valves must be kept closed to prevent water flowing into the bilge, but they should always be kept open when working with surface condensation.

Bilge suctions.—Circulating pumps, whether reciprocating or centrifugal, are generally fitted with an additional suction pipe leading to the bilge, with a non-return valve and strainer at its end, so that, in case of a serious leak, the water for condensation might be drawn from the bilge instead of the sea; and from the large quantity of water required for surface condensation, as pointed out in

Chapter XII., it may be readily understood how important an item this forms in the pumping arrangements of a ship. The bilge suction to the circulating pump should be as large as the delivery pipe in order to obtain the greatest possible pumping power in case of necessity. The centrifugal pump, which is worked by a separate engine, could be utilised for pumping out the ship if required, whether the main engines were at work or not, but the reciprocating pump can only be so used when the main engines are under way.

Bilge-injection.—In surface condensers ordinary bilge-injection valves and pipes are also frequently fitted, so that both air and circulating pumps may be used for bilge purposes in case of emergency. In some vertical single-acting air-pumps, a bilge suction is led to the air-pump instead of fitting a bilge-injection. The result will be the same in both cases.

Duplicate centrifugal pumps.—The present practice in war-ships is to fit, for each set of main engines, two centrifugal pumps and engines, each large enough to circulate all the water required for full power working. This provides for the case of accident to any circulating engine or pump and doubles the pumping power in the event of a leak. Each pump is fitted with separate sea and bilge suctions and also with a direct discharge overboard without passing the water through the condenser. The valves for changing from sea to bilge suction are arranged to be worked from the starting platform, and to enable this to be done quickly in case of emergency, the sea and bilge suction valves are coupled together so that they may be worked by a single lever. When this arrangement is carried out no sea or bilge injection valves and connections are fitted to the condensers.

Supplementary feed or charging cock.—In order to make up the loss of water that results from the safety

valves blowing, leakages, steam blast, exhaust steam from auxiliary engines (when they do not exhaust into the condenser), and other causes, a small cock, usually called a supplementary feed or charging cock, is fitted, by means of which water may be admitted to the hot-well from the circulating chamber to supply the feed-pumps. In some cases this cock delivers into the condenser, so that the additional water assists the condensation and is heated thereby before entering the hot-well.

Feed-tanks.—In surface-condensing engines the feed-pumps, instead of taking their supply from the hot-wells direct, usually draw from open wrought-iron tanks which are connected by suitable pipes to the hot-wells. One advantage of this arrangement is, that it insures a head of water being kept over the delivery valves of the air-pumps, which improves their working. The pipes connecting the hot-wells to the feed-tanks have cocks or valves fitted in them, so that when jet injection is used the quantity of water entering the feed-tank may be regulated. Gauge glasses and fittings similar to those on the boilers are usually fitted on the hot-wells or feed-tanks of surface condensers to show the level of the water in them.

Securing ends of condenser tubes.—In Figs. 131 and 132 the two methods most generally adopted of securing the ends of the condenser tubes are shown. In Fig. 131 all the tubes are packed at one operation, a sheet of india-rubber with suitable perforations being placed over the ends of the tubes, and a plate with corresponding holes fastened outside. This plan is objectionable on account of the difficulty of insuring the tightness of the joint around each tube; and if one tube leaks, the defect cannot be made good without removing the india-rubber sheet and covering plate from the whole of the tubes, and replacing them afresh.

The more general plan, therefore, is to pack each tube separately, as shown in Fig. 132. Small screwed stuffing-boxes are made in the tube plates, and screwed ferrules, fitting over the tubes, tighten tape packings around the tubes at the bottoms of the stuffing-boxes. The ferrules are generally made with small projections or flanges at their outer ends, to prevent the possibility of the tubes being forced out of place.

FIG. 131.

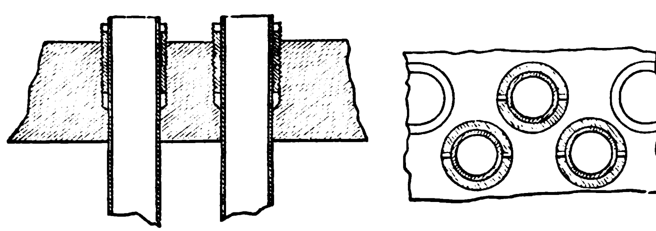


FIG. 132.

Vacuum gauge.—The vacuum in the condenser when the engines are at work is indicated by a Bourdon gauge, similar in construction to that shown in Fig. 45, which is generally called the vacuum gauge. This gauge is graduated to represent inches of mercury, and does not show, directly, the *absolute* pressure in the condenser, but only its relation to that of the atmosphere. Consequently, in judging of the performance of an engine, it is important that the state of the weather barometer should be noted.

The *actual pressure* in the condenser is independent of the atmospheric pressure, but the indication on the vacuum gauge is caused by the difference between the pressure of the atmosphere outside and the condenser pressure inside the tube of the gauge, so that the vacuum *registered* will vary almost directly with the atmospheric pressure. For example, suppose the constant pressure in the condenser to be represented by four inches of mercury. Then when the barometer stands at 30·5, the vacuum gauge would register $30{\cdot}5-4=26{\cdot}5$; whilst if the barometer stood at 29·5 the vacuum gauge could only indicate $29{\cdot}5-4=25{\cdot}5$ inches, or one inch lower than in the former case.

This may be readily understood by reference to an indicator diagram. We will assume the actual pressure in

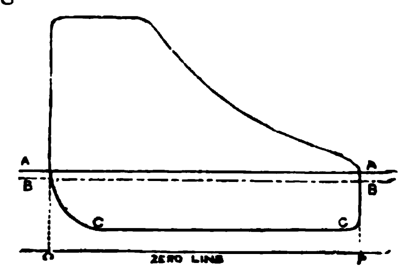


FIG. 133.

the condenser to be the same in each case. If the weather barometer stand at 30·5, the distance of the atmospheric line above the zero line O P, will represent a pressure due to 30½ inches of mercury, and is shown by the full line A A. If the barometer stand at 29·5, the atmospheric line will only represent a pressure due to 29½ inches of mercury, as shown by the dotted line B B; so that, the back pressure line C C being the same, it is evident that the part of the diagram below the atmospheric line will be reduced in a direct ratio with the height of the weather barometer. It should be observed that this does not affect the area of the diagram, but only the indicated vacuum.

The principal object secured in recording the actual height of the weather barometer is the determination of the maximum attainable vacuum in the condenser. To determine this the temperature of the condenser must be known, and the pressure of steam corresponding to this temperature ascertained by reference to tables, or by calculation. If this pressure be deducted from the atmospheric pressure given by the height of the barometer, the remainder will be the maximum attainable vacuum with that temperature of condenser. The difference between the actual vacuum indicated and the maximum attainable, will clearly be due to inefficiency in the action of the condenser, air-pumps, &c.

For example, if the temperature of the condenser be 100° Fahr., and the weather barometer stand at 30 inches, which is equivalent to a pressure of 14·7 pounds per sq. inch, we have;—pressure of vapour due to temperature of 100° Fahr. $= 0{\cdot}942$ pounds per sq. inch, and therefore the maximum vacuum attainable will be $= 14{\cdot}7 - 0{\cdot}942 = 13{\cdot}758$ pounds per square inch below the atmospheric pressure, which would be represented by about 28 inches on the gauge. If therefore the vacuum in the condenser, as shown by the gauge, be less than this it must be due to some inefficiency in the action of the condensers, air-pumps, &c.

Independent air-pumps.—In some recent engines, especially those designed to work at high speeds of revolution, the air-pumps, as well as the circulating, feed, and bilge pumps, have been arranged to be worked by separate auxiliary engines, so that the main engines would be employed for propelling purposes only. This simplifies the design of the machinery, and enables a vacuum to be always maintained in the condensers whether the main engines are working or not.

Air-pump valves.—Air-pump valves are usually made

of vulcanised india-rubber. This has to be specially prepared to resist the action of the mineral oil used for the lubrication of the cylinders and slide-valves, which soon destroys ordinary india-rubber. The material generally used for air-pump valves in naval ships contains as much as 70 per cent. of oxide of zinc in its composition, to enable it to resist the action of the oil.

Metallic valves.—In many cases valves made of thin sheet metal have been used in lieu of india-rubber valves

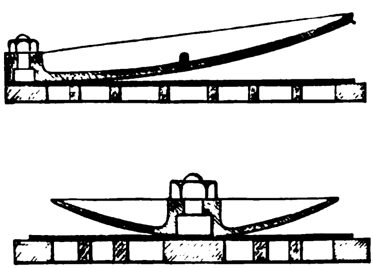


FIG. 134.

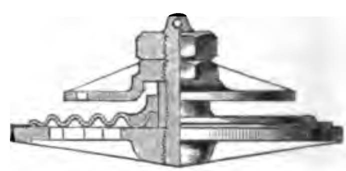


FIG. 135.

with satisfactory results. Coe & Kinghorn's patent valve consists of a thin flat plate of phosphor bronze, secured at its centre or at one end by a suitably curved guard to produce a gradual bending when the valve opens. This is shown in Fig. 134. Another form of metallic valve known as the *corrugated metallic valve* is shown in Fig. 135. This valve is made of sheet Delta metal and is in the form of a circular disc with annular corrugations. Both of these metallic valves work on gratings similar to those used with india-rubber valves.

CHAPTER XVIII.

ROTATORY MOTION.

We now proceed to describe the mechanism and fittings necessary for the transformation of the reciprocating motion of the piston into the rotatory motion of the shafting and propeller. This is performed by the agency of the connecting-rod and crank-shaft, and this step was a most important one in the progress of steam navigation. As pointed out in Chapter I., this motion, now so common

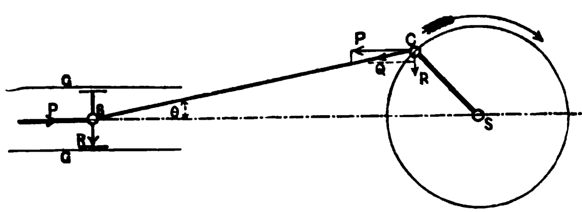


Fig. 136.

that it is little thought of, may be considered as the step that rendered steam navigation possible, and it will be advantageous to devote a few lines to its consideration.

The motion may be readily understood by reference to the outline diagram, Fig. 136.

The rotating shaft S, which is carried in suitable bearings, has on it a crank or arm S C, connected to the piston-rod by the connecting-rod C B, which has a working joint at each end. The end B of the piston-rod is constrained to move in a straight line by the action of suitable guides

G G. It is thus easily seen that the reciprocating motion of the piston is transformed, through the medium of the connecting-rod B C, into the rotatory motion of the crank-shaft S, from which the motion is communicated to the propeller.

The force P acting on the piston-rod, which we will assume to be constant, is opposed by the resistance offered to the revolution of the crank from the action of the propeller, and this produces a thrust, Q, in the connecting-rod. When the crank is at the dead point, the thrust on the connecting-rod is clearly equal to that on the piston. For any other position the thrust on the connecting-rod will be greater, increasing as the inclination is increased. This produces a reaction, R, perpendicular to the guides G G.

By applying the principle of the parallelogram of forces, it is easily seen that for any angle of the connecting-rod,

$$Q \cos \theta = P$$

$$Q \sin \theta = R$$

or,

$$Q = \frac{P}{\cos \theta} = P \sec \theta$$

$$R = P \tan \theta.$$

and

$$Q = \sqrt{P^2 + R^2}$$

If the direction of motion be as indicated by the arrow, the reaction on the guide is always downwards throughout the revolution. As a general rule marine engines are arranged so as to revolve in this direction when going ahead, and the lower guide surfaces are made larger than the upper surfaces, as the engines rarely have to work astern at full power for any great length of time. This point is especially important with the ordinary slipper or V guides, in which the lower rubbing faces are necessarily considerably wider than the upper faces. In Penn's trunk engines the motion is in the reverse direction, so that the

thrust of the connecting rod is upwards and tends to support the weight of the piston.

It is important to notice that no power is lost by the intervention of the crank. The motion of the crank is approximately uniform, that is, it passes through equal arcs in equal times, whilst that of the piston approaches harmonic motion, the velocity increasing from zero at the beginning to a maximum at the middle of its stroke, when retardation commences, and the velocity is again reduced to zero at the end of the stroke, This variation in the motion, however, does not affect the amount of work done, the rotative effect on the crank for *equal movements of the piston* being the same at all parts of the stroke, if the weight of the reciprocating parts of the engines be neglected, though the work done for *equal angles of the crank* will vary.

The weight of the reciprocating parts of the engine somewhat modifies the above statement, as it affects the distribution of the work throughout the stroke, the total amount of work, however, remaining unaltered. It is necessary to guard against the popular error of supposing that work is lost in consequence of bringing the masses of the pistons, rods, &c. to rest and starting them again in motion in the opposite direction twice in every revolution of the engine, and that there is a resultant loss of efficiency in reciprocating engines. The piston, rods, &c., during the first half of the stroke receive acceleration from the steam pressure on the piston, and the work thus accumulated is given out by pressure on the crank-pin during the retardation, so that nothing is lost, but the distribution of the work is only altered. This point is investigated more fully in the Appendix.

Connecting-rod.—In Figs. 137, 138, 139, and 140, the forms of connecting-rods most generally employed are shown. In Fig. 137, the connecting rod has T ends, and

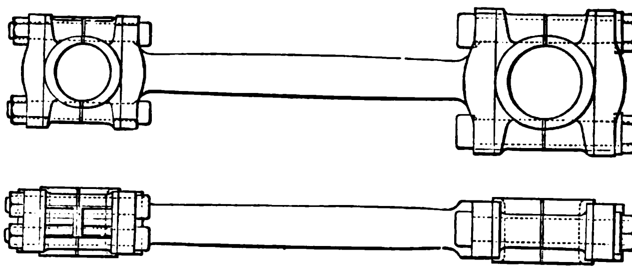


Fig. 137.

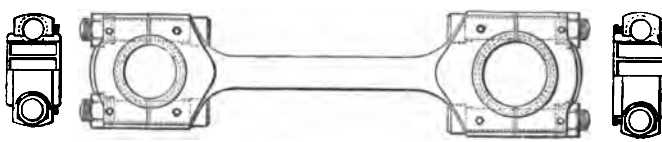


Fig. 138.

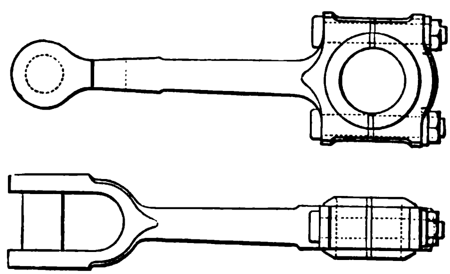


Fig. 139.

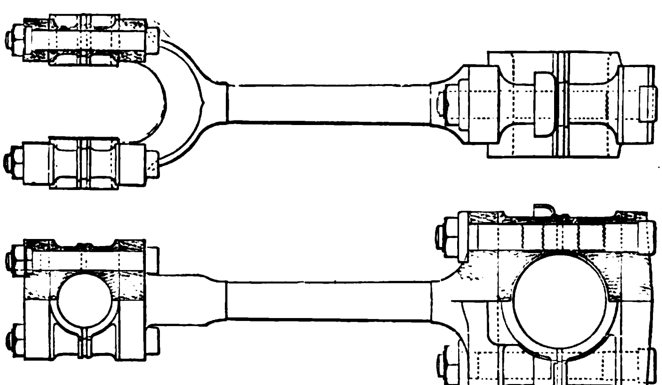


Fig. 140.

the brasses are secured, between the ends of the rod and strong wrought-iron or steel straps, by means of suitable bolts. In Fig. 138 one part of each of the heads is forged solid with the rod, the other parts being made of corresponding forms, so that cylindrical holes may be bored, to carry thin brasses as in an ordinary bearing. On the whole this form, with the semi-solid head, is the neatest and most efficient, especially for large engines, and it is most generally adopted in marine engines. Fig. 139 shows the form of connecting-rod usually adopted in direct-acting engines, one end being forked to fit the brasses on the T end of a piston-rod of the description shown in Fig. 113. In Fig. 140 the brasses are fitted on the forked end of the connecting-rod instead of on the end of the piston-rod.

The brasses of connecting-rods are not allowed to abut on each other, but have studs or thin liners placed between them, so that, as they wear, the liners or studs can be taken out and filed thinner, to allow the bearings to be adjusted. It is important that the brasses should be screwed hard on to the liners or studs, to insure correct working and to prevent straining and bending of the bolts. The nuts on the connecting-rod bolts are kept from turning by means of set screws, to prevent their slacking back when the engines are at work.

Crank-shafts.—Figs. 141 and 142 show crank-shafts of two-cylinder marine engines. Fig. 141 represents the complete shaft made in a single forging; but crank-shafts are now generally constructed in parts, connected together by flanged couplings, a separate part being provided for each cylinder, as shown in Fig. 142. This simplifies the operations of forging and turning, as there is only a single crank on each part, and the several pieces are made symmetrical, so that a spare crank may be carried, to be fitted in case of accident.

Balance weights.—Balance weights are sometimes

fitted on the cranks, as shown in Figs. 141 and 142, to counterbalance the weights of the crank-arms and connecting-rod heads. The pressure on the piston, however, is so enormously in excess of the weight balanced, that this refinement is found practically unnecessary to produce sufficient uniformity of motion; and as the balance weights are always inconvenient and may be a source of danger by the bolts loosening or breaking, their use is now being generally discarded. It is clear that in vertical engines it would be impossible to balance the reciprocating weights in this manner.

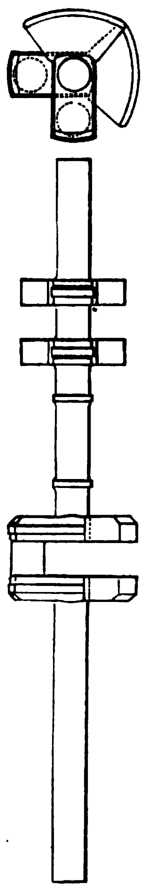


FIG. 141.

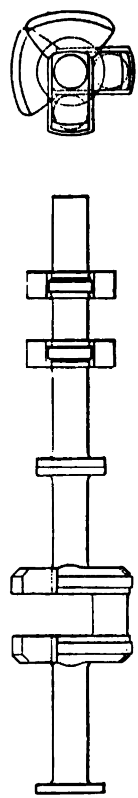
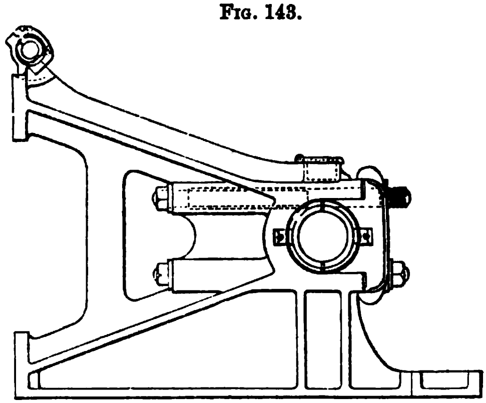


FIG. 142.

Turning wheel and gear.—On the after end of the crank-shaft a large worm wheel is keyed, which is fitted for the purpose of enabling the engines to be turned by hand when not under steam. The worm is generally worked by a ratchet and lever, which is a slow process in large engines. In most cases a small auxiliary engine is fitted to work the worm so that the engines may be turned more rapidly when under repair, for the fitting of connecting-rods, &c. When not under steam the engines should be moved a little daily by the turning gear, to keep them in good order.

FIG. 143.



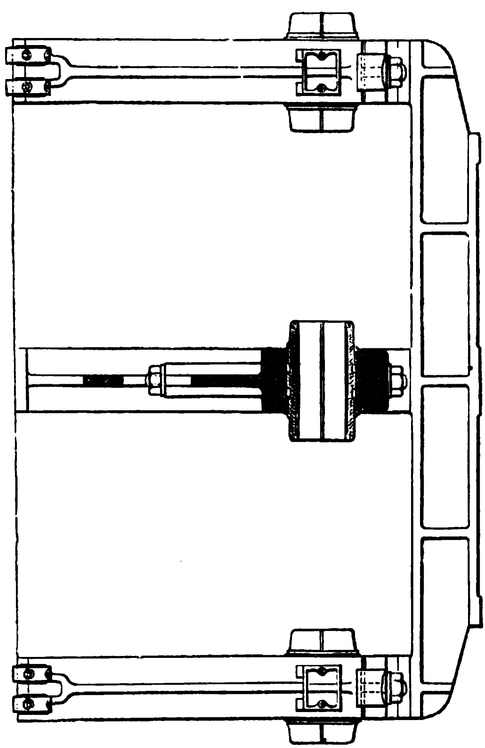


FIG. 144.

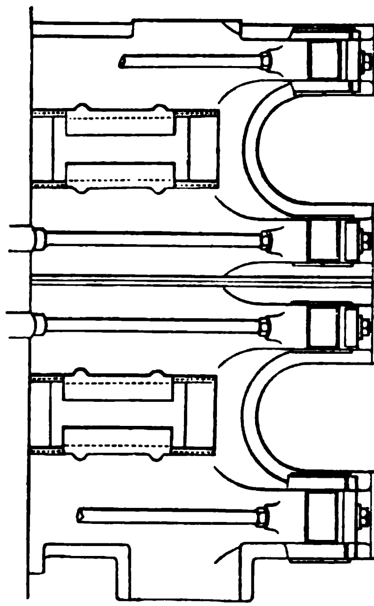


Fig. 146.

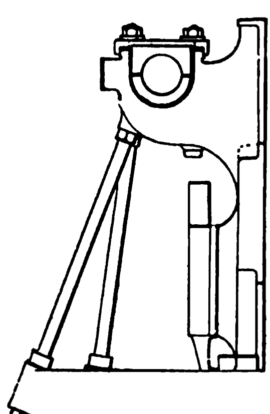


Fig. 145.

Main frames and bearings.—In horizontal engines the main bearings generally form parts of strong cast-iron frames rigidly attached to the cylinders and the engine-bearers. In recent ships, in order to reduce the weight of the machinery, the framing has, in many instances, been made of cast-steel. The more usual forms of main engine

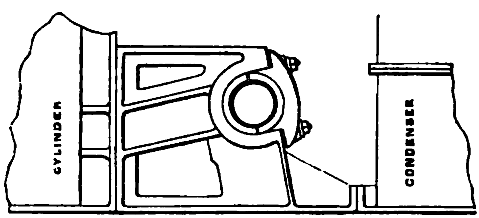


FIG. 147.

frames and bearings, by different makers, are shown in Figs. 143 to 147, from which the general arrangement and details of fitting can be easily seen. Fig. 148 shows the arrangement of cast-steel framing for H.M.S. 'Thames,' constructed by Messrs. John Penn and Sons.

In vertical engines the crank-shaft bearings are contained in the *sole* or *foundation plate*, which is rigidly secured to the hull of the ship by means of the engine-bearers. The cylinders are supported by standards or columns bolted to the sole-plate. In many vertical engines the condenser is fixed on the sole-plate, and the back supports for the cylinders are attached to the tops of the condensers and carry the guides for the ends of the piston-rods. This arrangement is shown in Fig. 149. In this case wrought-iron or steel columns are employed for the front supports of the cylinders, which is a convenient arrangement, as it makes all the working parts very accessible. In the engines of many mercantile steamers the front as well as the back standards are made of cast-iron, and the end of the piston-rod is attached to a cross-

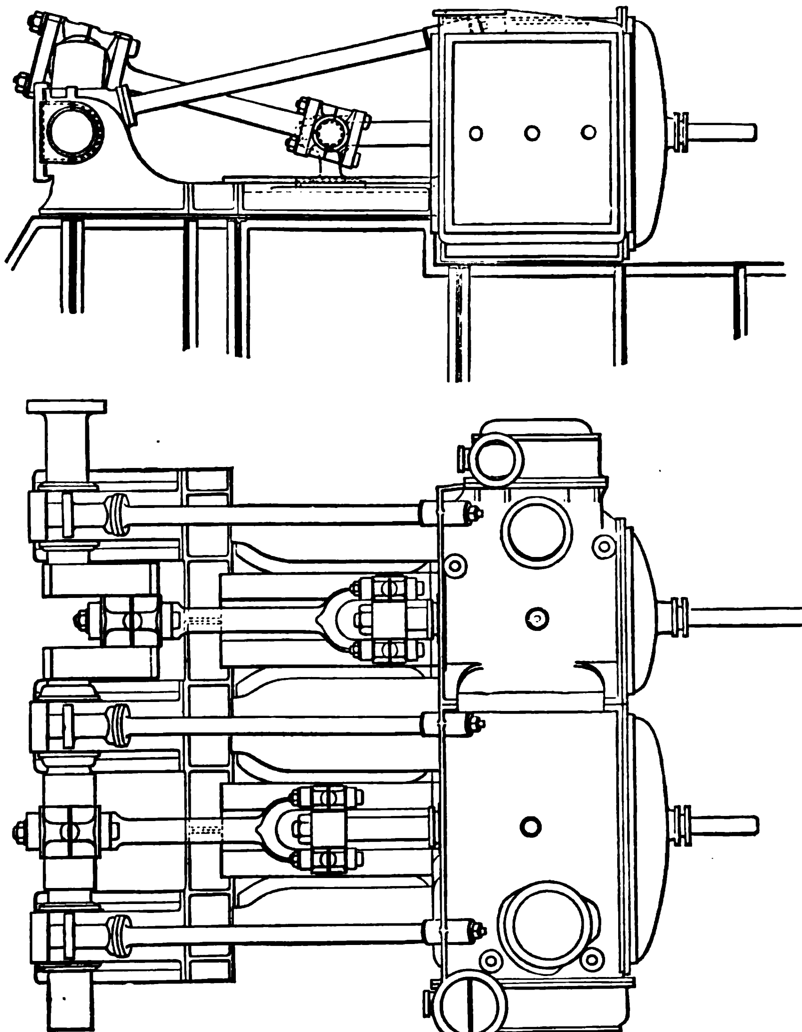


Fig. 148.

head that works between guides carried on the front and back standards. This forms a very strong and rigid arrangement, but it has the disadvantages of being much heavier and of obscuring the working parts and rendering

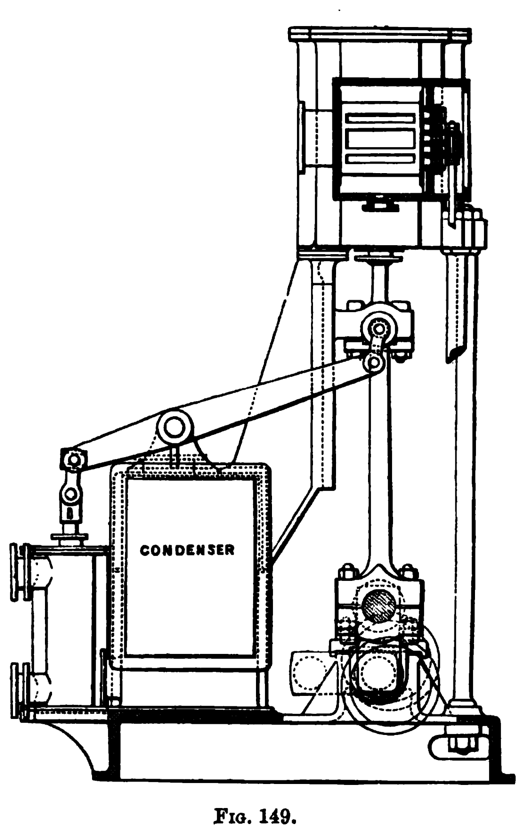


Fig. 149.

them less accessible for examination and repair. In many war-ships the framing is entirely constructed of wrought-iron or steel suitably trussed to give sufficient rigidity. The system carried out by Messrs. J. and G. Rennie in the

engines of H.M. twin screw armour-clad ship 'Hero,' of 6,000 I.H.P., is shown in Fig. 150.

The brasses in all main bearings should be so fitted as to allow the back or bottom brass to be removed for examination when necessary, without involving the operation of first taking out the crank-shaft. To effect this, the brasses are usually made concentric with the shaft, and, in horizontal engines, are held in position by small guard plates bolted to the sides of the frame, and fitting into recesses

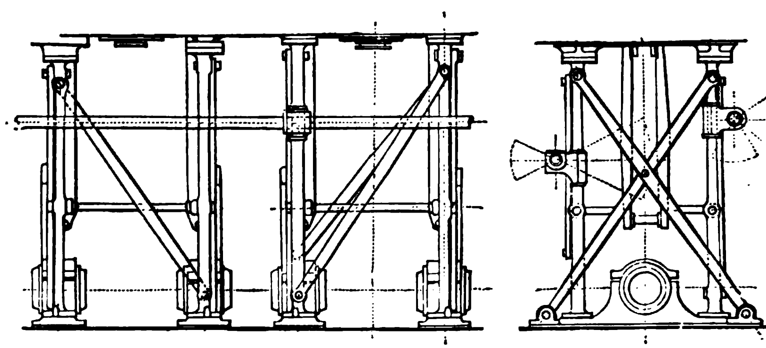


Fig. 150.

cut in the flanges of the brass. When the cap and outside or top brass are removed the back or bottom brass may be taken out by revolving the brass round the shaft.

White metal bearings.—Most of the bearings in marine engines are filled with a soft white metal made of tin, antimony, lead, &c., with a small percentage of copper. The compositions used vary considerably, as shown in the following analyses of metal from bearings that have worked satisfactorily.

	1	2	3	4	5
Tin	10	16	41	63	32
Copper	1	2	0·25	2	4
Antimony	1	2	11		—
Lead	—	—	48	35	1·1
Zinc	—	—	—	—	62
Iron	—	—	—	—	0·6

No. 1 is Babbitt's metal. Nos. 2 and 3, though differing so widely in composition, have each given very good results in practice.

These alloys are soft and plastic, and if they be well lubricated will sustain a great pressure without heating. The white metal is confined by fillets or rims cast on the bearings, to prevent its being squeezed out, the depth of the recesses for the metal being usually from $\frac{5}{16}$ to $\frac{3}{8}$ inch. In some cases the white metal is fitted in the form of discs or plugs, and sometimes as strips dovetailed into the bearings in the manner adopted for the lignum vitæ bearings for stern-fittings.

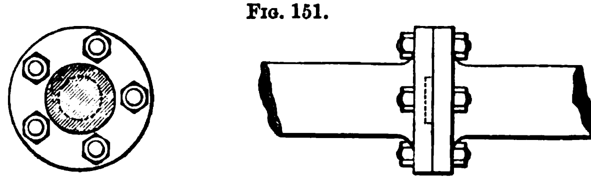
Propeller-shafting.—From the crank-shaft the rotatory motion is communicated to the propeller by means of the screw-shafting, which consists of straight lengths of shafting of somewhat smaller diameter than the crank-shaft, as they only have to resist torsion and are not exposed to bending action. The couplings for securing the different lengths of the shafting together are shown in Figs. 151 and 152.

Plummer block.—The propeller-shafting is carried by plummer blocks of the form shown in Fig. 153, which are usually made of cast-iron and lined with white metal. These bearings should not be farther apart than twenty times the diameter of the shaft.

Thrust block.—On the foremost length of the propeller-shafting a bearing of somewhat peculiar form is usually fitted, to receive the thrust of the propeller, and transmit

it to the ship. In order to reduce the intensity of the friction, the thrust journal on the shaft is made with

FIG. 151.



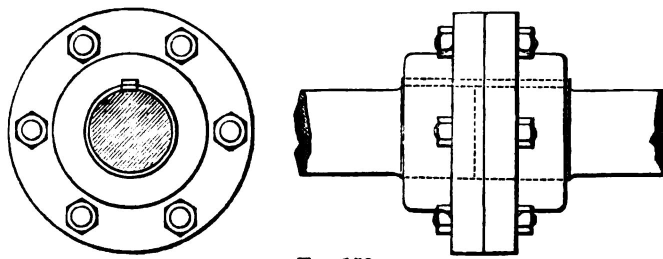


FIG. 152.

several collars, which press on properly fitted thrust surfaces in the bearing. Fig. 154 shows an arrangement now

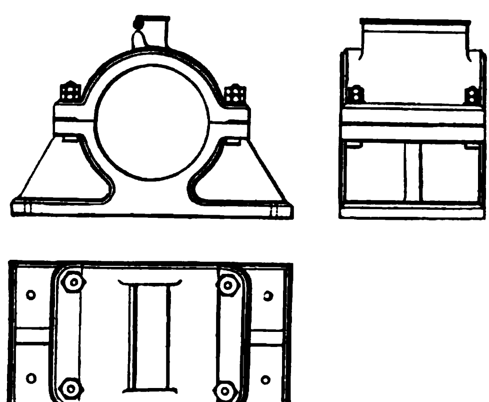


FIG. 153.

often adopted, in which the rubbing surfaces of the block are made of white metal.

The block shown in the sketch has an annular space round it through which a stream of cold water is kept running when the engines are at work, in order to keep the bearing cool. It is not, however, clear that any advantage that may be gained by this is sufficient to compensate for the increased weight and cost of the casting required, and it is possible that heating of the bearing would not be so readily detected by the touch, as the outer surface would always feel cold, which might cause incorrect infe-

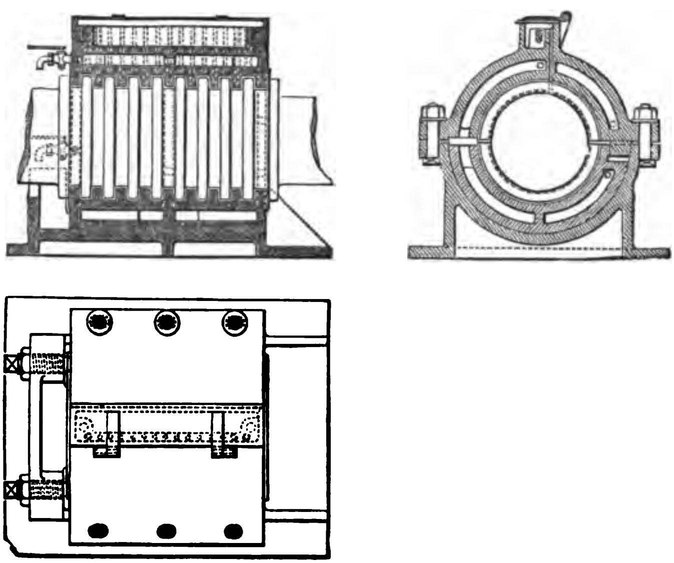


FIG. 154.

rences to be drawn as to the state of the rubbing surfaces, and the water chamber is not now fitted. In Fig. 155 separate brass thrust rings are fitted in the bearing to form the rubbing surfaces. These are made in halves and arranged so that they may be readily renewed when necessary. Fig. 156 shows a description of thrust-bearing that has been extensively used by Messrs. Maudslay, Sons, &

Field and some other engineers, which permits the position of the thrust rings to be adjusted, as required, by means of set-screws. The front view of the movable thrust-collars is shown at A.

Thrust blocks are carried on strong bracket plates secured to suitable bearers fixed to the ship, and set-screws

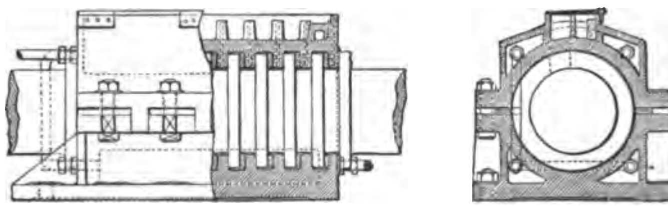


Fig. 155.

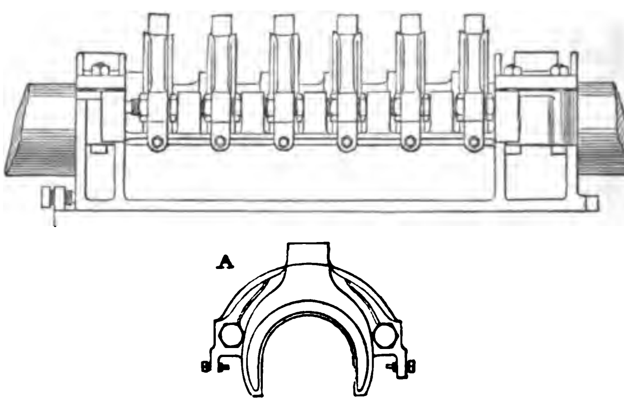


Fig. 156.

are fitted to the brackets to enable the position of the thrust blocks to be adjusted, within certain limits, as may be required. The holes in the base of the thrust block, for the holding-down bolts, are made elongated to admit of this. An ordinary plummer block should always be fitted close to the thrust-bearing to carry the weight of

the shaft, so that the thrust-bearing will only have to sustain the thrust of the propeller.

In recent single-screw ships the thrust of the propeller is taken on a disc on the stern-post fitted with lignum-vitæ segments. This plan, which is more fully described in Chapter XXIII., has proved very successful.

Driving coupling.—The propeller-shafting, in some cases, especially in wood ships, is not rigidly connected with the crank-shaft, but is driven by pins or studs in the after-coupling of the crank-shaft, as shown in Fig. 157. There is a small space between the two shafts, to prevent the thrust of the propeller being transmitted to the crank-shaft as the

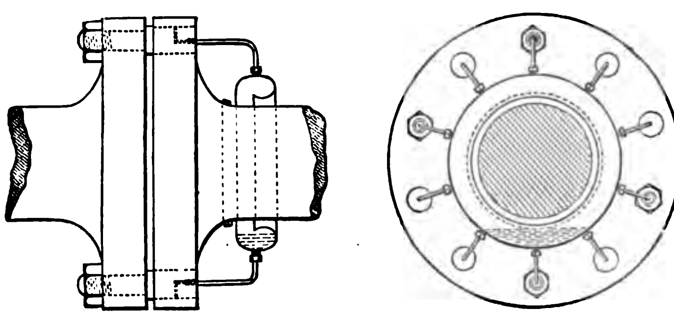


FIG. 157.

thrust-bearing wears, which would cause pressure and probably heating at the after-ends of the journals. Usually two or more of the driving bolts are fitted with nuts so that the crank-shaft may be drawn back if necessary to relieve the after collars. These driving bolts should be fitted with suitable lubricating arrangements to prevent their setting fast. A simple method of doing this is shown in Fig. 157, the oil being forced out to the bolts by centrifugal action as the shaft revolves.

Centrifugal lubricators for crank-pins.—A similar plan has been adopted in recent ships for lubricating the crank-pins. This is shown in Fig. 158. The crank-pins are

hollow, and small holes are bored from the rubbing surface to the central space, as shown by the dotted lines. An annular lubricator, A, on the crank-shaft is connected to the hole in the centre of the crank-pin, and by the revolution

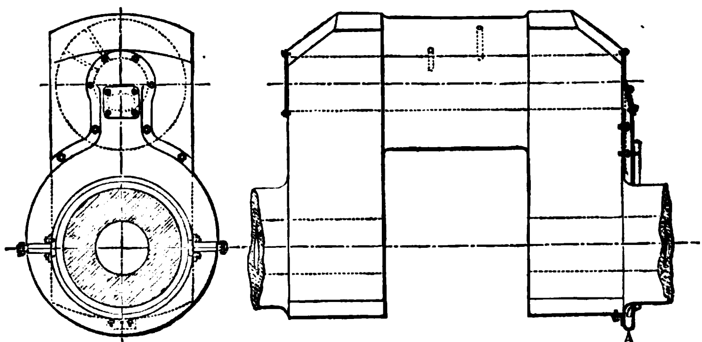


FIG. 158.

of the shaft the oil is forced out to the rubbing surface of the crank-pin. This system has been attended with most satisfactory results, and is now generally applied to all high-speed engines.

CHAPTER XIX.

DETAILS OF COMPOUND AND TRIPLE EXPANSION ENGINES.

IN the progress of steam navigation, when the fact was fully accepted that in order to make long voyages remunerative the pressures and rates of expansion of steam should be increased so as to reduce the expenditure of coal, the question of the strains brought on the framing and shafting of the engine by working the steam at a high rate of expansion in a single cylinder, became one of great importance; as, in large engines especially, the variation of pressure during the stroke would be so great that the maximum strains produced would probably be dangerous to the structure. Attention was consequently directed to the employment of the combined or compound engine, in which the high-pressure steam acts on a small piston only, and a reduced pressure on the large piston, so as to reduce the maximum strains on the framing, &c., and make the turning moments more uniform. As the working pressures increased additional stages in the expansion became necessary, and this led to the triple and quadruple expansion engines now used.

The only difference between the mechanism of compound and simple expansion engines is in the arrangement of the cylinders. All the other parts are the same in the compound engine as in any ordinary surface condensing engine. In the simple expansion engine, the

steam enters each cylinder direct from the boilers, and at the end of each stroke is exhausted direct into the condenser. In the compound system the steam from the boilers is only admitted direct to the smaller cylinder, usually known as the high-pressure cylinder, and at the end of the stroke in that cylinder, instead of passing direct to the condenser, the steam enters one or more additional cylinders, called the intermediate and low-pressure cylinders, in which the expansion is completed, after which the steam passes to the condensers. The boilers are only in direct communication with the high-pressure cylinders, and the condensers with the low-pressure cylinders.

The two principal types of ordinary compound engines, so far as the arrangement of the cylinders is concerned, are :—

1. With the high- and low-pressure cylinders in the same line and acting on the same crank; the two pistons being on the same rod, and the steam exhausting directly from the high- to the low-pressure cylinder. This type is sometimes known technically as the *tandem engine*, and its general arrangement is shown in Fig. 159. So far as the distribution of the steam is concerned, it is immaterial at which end of the low-pressure cylinder the high-pressure cylinder is placed, this point being settled solely by considerations of practical convenience.

2. With the cylinders side by side, operating on different cranks at various angles with each other. The steam in this case, on leaving the high-pressure cylinder enters an intermediate receiver or reservoir before entering the low-pressure cylinder.

The type shown in Fig. 159 has certain advantages and has been largely adopted by some engineers, especially for engines of great power, as with this form, consisting of two pairs of cylinders, the high-pressure being either

directly above or below the low-pressure cylinder, with the pistons on the same rod, large powers may be obtained without introducing castings of extraordinary size or complexity. This type of engine is easily handled, and

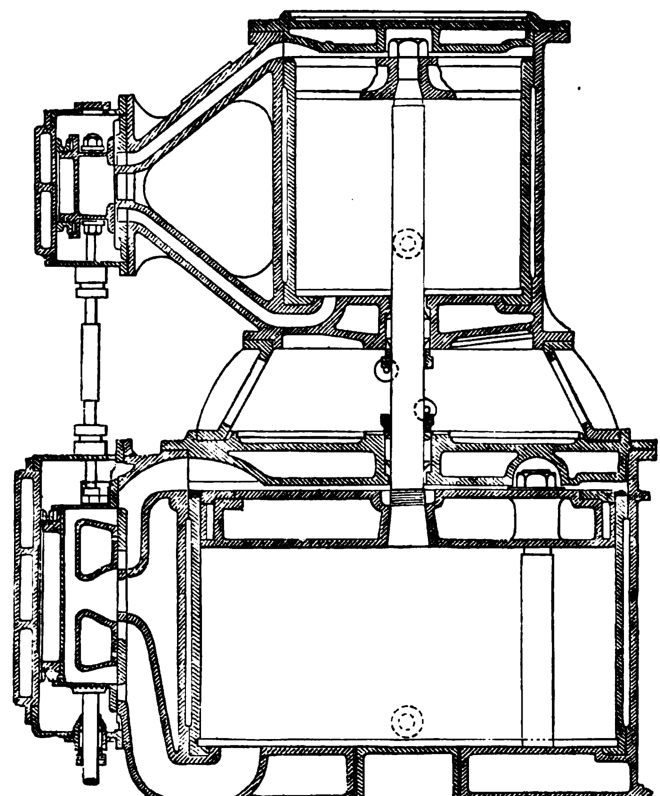


FIG. 159.

provides for an almost perfect expansion of the steam. It is also the readiest form to which an engine of the old type can be converted.

In Fig. 159 the high-pressure cylinder is shown above

the low-pressure cylinder. To the cover of the low-pressure cylinder is attached a bracket or distance piece for carrying the high-pressure cylinder, so that the piston-rod, common to the two cylinders, has a separate stuffing-box in the cover of each. In some of the earlier engines of this type, the two cylinders were separated only by a diaphragm with a bush in its centre, through which the piston-rod worked. This, however, was found to be unsatisfactory in practice, as when the bush wore, it allowed leakage from one cylinder to the other; this defect was difficult to remedy in consequence of its position, and the economy and efficiency of the engine were thereby reduced. In this type of engine the clearance spaces in the high-pressure cylinders are very great. The slide-valves of the high- and low-pressure cylinders are usually in the same line and worked by the same eccentrics, which necessitates long passages in the high-pressure cylinder, as shown in Fig. 159. This large clearance space decreases the expansive efficiency, and causes considerable waste of steam. This subject is more fully discussed in Chapter XXIV.

Messrs. Maudslay, Sons, & Field, in H.M.S. 'Sirius,' and in the fast despatch vessels 'Iris' and 'Mercury,' have fitted the high-pressure cylinder in front of the low. This was mainly due to the small amount of space that could be allowed for the machinery, and on the whole it is not so convenient an arrangement for general purposes as that shown in Fig. 159. A section showing this arrangement of cylinders is given in Fig. 160.

The arrangement of compound engines that was most generally adopted by engineers and steam users, for moderate powers, was one in which the high- and low-pressure cylinders were placed side by side, and the steam acted on cranks at right angles to each other; the exhaust orifice of the high-pressure cylinder being connected by a pipe to the low-pressure slide-valve casing.

In some of the earlier engines of this class, unnecessary complexity of construction was introduced, by casting a large reservoir around the high-pressure cylinder, which was considered essential to receive the steam on its exhaust from the high-pressure cylinder previous to its admission to the low-pressure cylinder. When the cylinders were jacketed as well, which was generally the case, the cylinder, jacket, and reservoir being made in one piece, formed a very

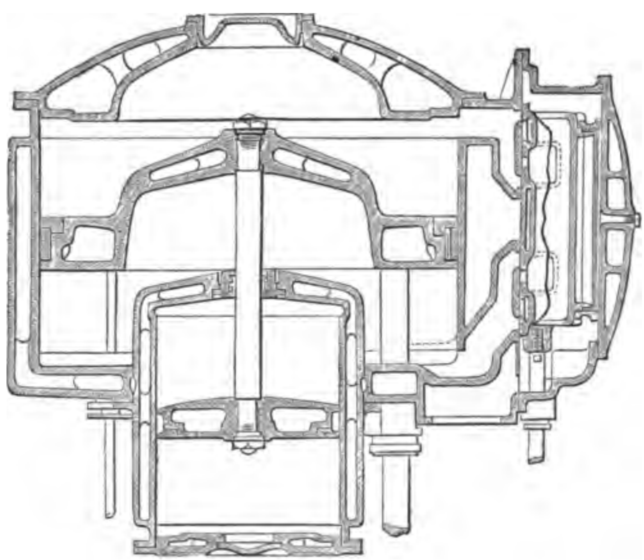


FIG. 160.

heavy, difficult, and expensive casting; and the unequal contraction of the several parts on cooling frequently caused the metal to crack in various places, and gave trouble in many other ways. An example of this kind is shown in Fig. 161.

A large intermediate receiver of this description is, however, not required; and it is found that all that is necessary is an ordinary exhaust pipe from the eduction

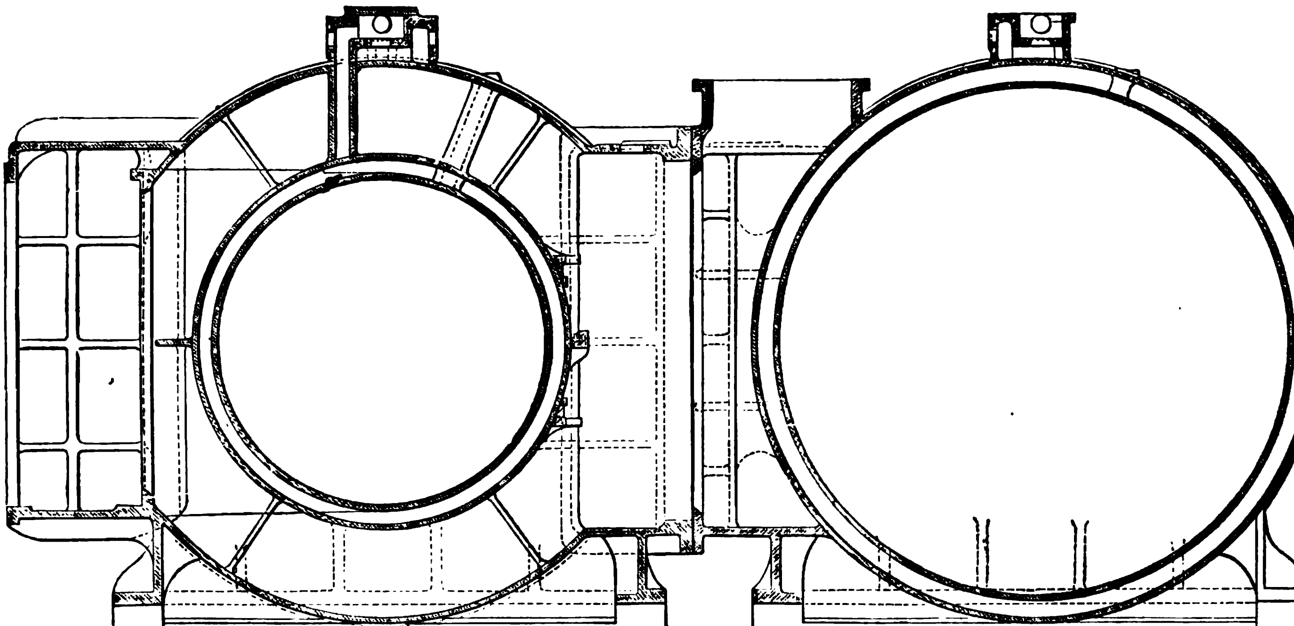


FIG. 161.

orifice of the high-pressure cylinder to the steam inlet of the low-pressure cylinder, the volume of the exhaust passage of the high-pressure cylinder and of the intermediate pipe and low-pressure valve casing being quite sufficient to allow for the compression that takes place between the release from the high-pressure cylinder and admission to the low-pressure cylinder; and most modern engines have been made in this way with very satisfactory results. With this method of construction, the cylinder castings for the compound engine are quite as simple as for an ordinary expansion engine; the only difference in the arrangement being that the exhaust pipe from the high-pressure cylinder is led to the slide-valve casing of the low-pressure cylinder, instead of to the condenser. In point of fact the cylinder castings are independent of each other, and precisely the same, except in size, as if they had been made for a simple expansion engine.

The capacity of the intermediate receiver is immaterial so far as the power of the engines is concerned, its only effect being shown on the back pressure line of the high-pressure diagram, which becomes more nearly straight, as the volume of the receiver is increased. It is probable, however, that the larger the receiver the greater will be the loss of efficiency from radiation and liquefaction.

Three-cylinder engines.—The ordinary three-cylinder compound engine is simply a modification of the type just described; but instead of a single low-pressure cylinder, two cylinders are used, the steam on exhausting from the high-pressure cylinder to the receiver operating on the pistons of the two low-pressure cylinders, as simultaneously as the relative positions of the low-pressure cranks will admit, both cylinders being in communication with the same intermediate receiver.

By the term *receiver* is to be understood the whole of the space between the high-pressure piston and the back

of the low-pressure slide-valve or valves, comprising the volumes of the steam and exhaust passages of the high-pressure cylinder, the pipes from the high-pressure cylinder to the low-pressure valve casings, and the low-pressure valve casings themselves. This will, perhaps, be more clearly understood by reference to the Appendix, in which the action of the steam through the cylinders of compound engines is traced, and the methods of drawing the theoretical diagrams explained.

The three-cylinder type of compound engine is used when the power is so great that the employment of a single low-pressure cylinder would be inexpedient on account of its unwieldy dimensions, so that the division of the work between two low-pressure cylinders is preferable. The power of a compound engine, working at any given rate of expansion, depends entirely on the dimensions of its low-pressure cylinder or cylinders, and is not affected by the size of the high-pressure cylinder, which must only be regarded as a stepping-stone in the expansion. The capacity of the low-pressure cylinder or cylinders of a compound engine requires to be the same as that of the whole of the cylinders of a simple expansion engine of the same power working at the same initial pressure of steam and total ratio of expansion. This is easily seen from the consideration, that since the initial pressures and the ratios of expansion are the same, the final pressures and volumes must be identical in the two cases. In the simple engine the whole of the steam at the end of the expansion fills all the cylinders, whilst in the compound engine it is contained by the low-pressure cylinders only. Consequently the capacity of the low-pressure cylinders of the compound engine must be equal to the capacity of all the cylinders of the simple expansion engine. The necessity of having two low-pressure cylinders instead of one in the case of large powers is therefore obvious.

In the majority of three-cylinder compound engines the high-pressure cylinder is fitted between the two low-pressure cylinders, but in many cases it is placed forward or aft of the two low-pressure cylinders. The relative positions of the cylinders is immaterial so far as the distribution of the steam is concerned, and is determined solely from reasons of practical convenience.

The angles at which the cranks are placed with respect to each other are very varied. In the majority of cases they are set at equal angles of 120° with each other, as shown in Fig. 162 A; but some makers consider that better results, either in the direction of improved distribution of the steam, or of uniformity of twisting moment, may be obtained by placing the cranks at different angles with each other.

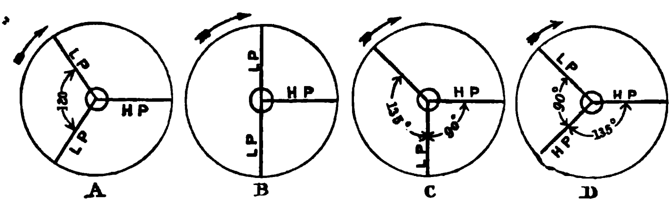


FIG. 162.

Messrs. J. & G. Rennie, in the 'Boadicea' and 'Bacchante,' placed the cranks of the low-pressure cylinders opposite to each other, both being at right angles to the high-pressure crank, as shown in Fig. 162 B; which arrangement they considered would reduce the maximum strains on the machinery, and so produce the greatest regularity in the turning moment. Messrs. Humphrys, Tennant & Co., with the object of improving the distribution of the steam between the two low-pressure cylinders when working at the maximum power, have in several ships placed one low-pressure crank at right angles to the high-pressure crank, and the second low-pressure crank at

an angle of 135° with each of the other two, as shown in Fig. 162 C. In many French engines the low-pressure cranks have been placed at right angles to each other, and the high-pressure crank made an angle of 135° with each, as shown in Fig. 162 D. It is, however, doubtful if any of these variations possess any practical advantage over that of placing the cranks at equal angles with each other, which, with suitable valve arrangements, will probably be found to be the most convenient in the majority of circumstances.

Messrs. John Penn & Sons, in H.M.S. 'Northampton,' 'Ajax,' 'Agamemnon,' and several other vessels, have fitted a *composite* type of engine which is capable of being worked either with simple or compound expansion as may be desired. It consists of three cylinders of equal size, acting on cranks placed at equal angles with each other, and when working at full power all the cylinders take steam direct from the boilers and exhaust into the condensers. Special valves are fitted in the steam and exhaust pipes, so that when reduced powers are required, the foremost cylinder, only, takes steam from the boilers, and exhausts into the steam pipe leading to the other two cylinders, which form the low-pressure cylinders of the arrangement and exhaust into the condensers. By this plan the dimensions of the cylinders are reduced, and the spare gear is simplified, as compared with the ordinary three-cylinder compound engine. It also provides a ready means of working with low-pressure steam when required, and the resultant turning moments on the crank-shafts are very uniform; but it is considerably less economical than the compound engine at full power, and does not appear to be more so at reduced powers, when both engines are working under similar conditions.

The great advantage of the compound engine, so far as its mechanism is concerned, is the facility with which

it allows high rates of expansion of steam to be carried out without bringing excessive strains on the framing. For example, if we consider the cases of two engines, one simple, the other compound, each working with 60 pounds initial pressure of steam, and developing 2,100 indicated horse-power, with a total rate of expansion of 8 times, we shall find that whilst the maximum turning moment in the case of the compound engine is 960 inch-tons, it would be 1,250 inch-tons in the engine with simple expansion, or more than 30 per cent. greater, the mean moment being the same in the two engines. In consequence of the greater uniformity of twisting moment, the shafting and framing may be made lighter in the compound than in the non-compound engine, and much greater steadiness of motion may be obtained, and more efficient action of the propeller in the water expected. The great variations of pressure to which the shafts of simple engines are exposed, when worked at high rates of expansion, appear to produce the same effect on the material that vibration does, viz., to cause the structure to become crystalline. Several cases of broken shafts have occurred in engines of this class, and the result was considered to have been due to the excessive intermittent strains brought on them.

There are practical difficulties in the arrangement of the mechanism for carrying out high rates of expansion with the non-compound engine, when reduced powers only are required, which do not exist in the compound engine. For example, in the case referred to above, if we take the stroke of the engines to be 33 inches, the steam would be carried in the high-pressure cylinder of the compound engine, when working at full power, for 14 or 15 inches; so that there would be plenty of range for the expansion valve to be used when working at low powers, and the pressure of steam in the boilers may be kept at or near its maximum. In the simple expansion engine the

steam would only be carried about 4 inches in the cylinders when working at full power, so that the practical range of expansive working is much reduced, and, when the engines are required to work at low powers, either the pressure of the steam in the boilers must be reduced, or the steam must be throttled, so that the benefits of high pressure and expansion are lost.

In the compound engine, although the steam is expanded 8 times when developing full power, it can be expanded many more times when working at reduced powers, whereas in the non-compound engine it cannot be expanded much more than 8 times, whatever the reduction in the power may be. This results from the necessary mechanical arrangements, and is altogether independent of any loss of efficiency that would ensue from liquefaction, &c., when attempting to carry out a high rate of expansion in a single cylinder. It is therefore clear that at reduced speed, which is the ordinary condition of service in warships, the compound engine would in general be more economical than the simple expansion engine.

The superiority of the compound engine would be more completely demonstrated as the engines became worn. When the slides and pistons began to leak, the loss in the simple engine would be much greater than in the compound, in consequence of the greater range of pressure in the cylinders of the simple engine. The steam leaking past the piston in the simple engine, would go direct to the condenser without doing any useful work, and would also increase the back pressure, whilst in the compound engine, the steam leaking past the high-pressure piston would have a chance of doing useful work in the low-pressure cylinder before getting to the condenser, and the amount of leakage in the low-pressure cylinder would be reduced on account of the smaller difference of pressure on the two sides of the piston in that cylinder.

Since the cylinders of compound engines provide in themselves for a considerable amount of expansion, special cut-off or expansion valves have in many cases been dispensed with in these engines, thus reducing the complexity and number of parts as compared with the non-compound engine, in which expansion valves suitable for early cut-off are an absolute necessity when high-pressure steam is used. By proper arrangement and setting of the slide-valves, the ordinary proportions of cylinders used in compound engines would give an expansion of from five to eight times at full power without any additional gear, and the increased expansion required for reduced powers could be obtained by the use of the link. This plan has the additional advantage of regulating the point of cut-off in the low-pressure cylinder so as to cause it to do a fair proportion of work at reduced powers.

Most compound engines, however, have been fitted with expansion valves on the high-pressure cylinder, and many have expansion valves also on the low-pressure cylinders. The expansion valve on the low-pressure cylinder, though it does not affect the total power developed by the engines, provides a simple and ready means of regulating the amount of work done by the two cylinders, when necessary, to equalise the strains on the machinery. Without this valve, if the engines be working at very low powers, as is the case in war-ships on ordinary service, it would be found, in many cases, that the work done in the low-pressure cylinder would be very small. By setting the low-pressure expansion valve to an early cut-off, the pressure in the receiver, which forms the back pressure in the high-pressure cylinder, would be increased, so that the work done in that cylinder would be diminished and that in the low pressure increased, and the power would consequently be more equally divided between the two cylinders.

These expansion valves are valuable in the case of

three-cylinder compound engines. By their use the total work can be equally divided between the three cylinders, and the twisting moments rendered practically uniform. Without them, the high-pressure cylinder would perform, generally, about one-half the total work developed by the engines, causing the pressure on the high-pressure crank-pin to be practically double that on either of the other two crank-pins. In the more recent engines, instead of fitting separate expansion valves to each cylinder, the length of the reversing arm has been made adjustable to enable the degree of expansion to be varied, as is explained at page 230. For further information on these points we must refer to the Appendix, in which the construction of indicator diagrams of compound engines is explained.

Auxiliary or starting valves.—To facilitate the handling of compound engines, in which the boiler steam pressure is admitted to one cylinder only, small auxiliary or starting valves are fitted to the cylinders, worked by levers from the starting platform. These valves should take their steam from a branch on the main steam pipe, and they provide the means of admitting steam, by hand, to the top or bottom of either of the cylinders, as may be necessary, to assist the starting of the engines. In some compound engines a single starting valve is fitted, to admit steam direct from the main steam pipe to the reservoir or low-pressure slide-valve casing; this is useful when the high-pressure crank is on the dead centre.

Change or converting valves.—In some ships in the Royal Navy, valves have been fitted to the high- and low-pressure cylinders, so that in case of necessity the engines may be worked as simple engines. These valves provide the means of connecting the high-pressure cylinder with the condenser and the low-pressure cylinder with the main steam pipe, if required. These fittings have been supplied, so that, if deemed desirable in action, the engines

may be worked with steam at or near the atmospheric pressure, to prevent disaster in the event of a shot striking the boilers or steam pipes. It is, however, necessary to maintain a pressure of at least 10 or 12 pounds above the atmosphere to keep the engines under control if the vacuum should fail, and with this pressure compound engines can be as readily manipulated as simple expansion engines. It is very doubtful if any naval captain would reduce the available power of his machinery, on which the safety of his ship would largely depend, on going into action. Most officers would prefer to keep up the steam pressure, in order to have the engines thoroughly under control, and the full power and speed of the ship at command for manœuvring, &c. No action can be fought without risk, and its casualties must be provided against, not by reducing the efficiency of the ship by working with steam at or about the atmospheric pressure, but by thoroughly protecting the vital parts of the ship, of which the region of the engine and boiler rooms is not the least important, and by subdivision of the engine and boiler rooms into separate compartments, so as to reduce the chances of disaster to a minimum. These change valves have not been fitted to the more recent compound engines.

Triple expansion engines.—The same arguments that have been employed to prove the superiority of the two-cylinder compound engine over the simple expansion engine when the working steam pressures were increased from 30 to 60 pounds per square inch, are applicable to the proof of the superiority of the triple expansion engine over the two-cylinder compound engine for the steam pressures now generally used. The essential argument in each case is the reduction of the variation in temperature of the cylinders during each stroke, which decreases the loss from liquefaction.

Some of the forms in which the triple expansion

system has been carried out are illustrated in Fig. 163. In Fig. 163 A, each of the cylinders is fitted over a separate crank, the high, intermediate, and low-pressure cylinders being arranged in succession. In the engines shown in Figs. 8, 9, and 10 the high-pressure cylinder is placed between the intermediate and low-pressure cylinders. For large powers the low-pressure cylinder

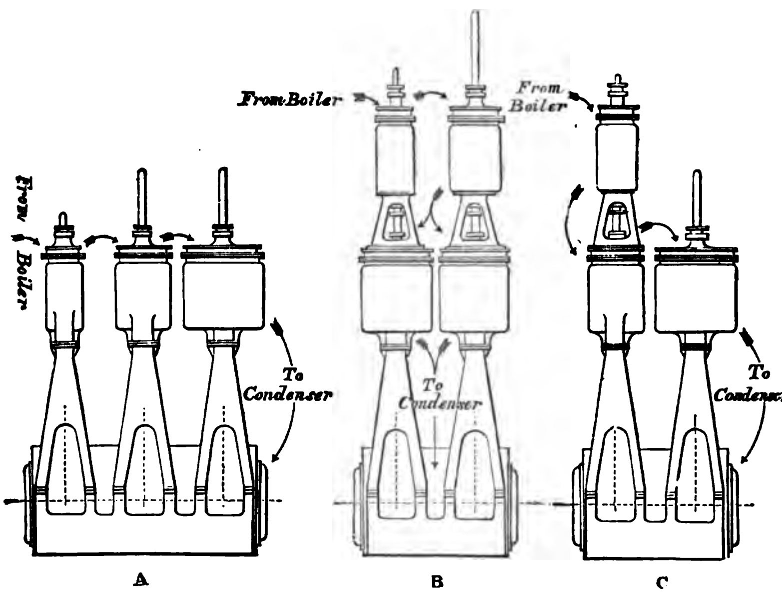


FIG. 163.

would require to be divided into two parts, from the same considerations that led to the three-cylinder compound engine, as explained at page 338. This would necessitate four cranks and an increase in the length of the engine-room. Fig. 163 B shows another form suitable for high powers which does not require a great length of engine-room, and produces fairly uniform strains on the shafting, the high and intermediate pressure

cylinders being above the two low-pressure cylinders. The arrangement shown in Fig. 163 C, though convenient in some cases, cannot be considered altogether satisfactory, as the strains on the crank-pins would be very unequal. It is easy to see the many different ways in which the several cylinders could be arranged to suit special cases. As previously pointed out the relative positions of the cylinders do not affect the distribution of the steam, and are settled entirely by practical considerations.

We will conclude this chapter by appending a specimen specification of the principal points to be considered in the design of the machinery for a modern ship of war.

Specification of certain Particulars to be observed in the Construction of the Machinery for a Twin Screw Armour-clad Ship of 12,000 *Indicated Horse-Power.*

1. *Indicated horse-power*—Each screw is to be driven by an independent set of engines, with three vertical inverted cylinders, of the collective power of 6,000 horses, giving an aggregate indicated power of 12,000 horses for both sets of engines. The machinery will be required to develop and maintain this power on the official trial for four consecutive hours at sea. A further trial at sea of four consecutive hours' duration is to be made without closing the stokeholds, during which it is expected that about 8,000 I.H.P. will be maintained. The total weight of the engines and boilers, including the water contained in them, and all other articles to be supplied under this contract, is not to exceed 1,100 tons. This weight is to be exclusive of the items in paragraphs 64 to 70.

2. *Cylinders.*—The diameters of the high-pressure cylinders are to be about 43 inches, of the intermediate cylinders about 62 inches, and of the low-pressure cylinders not less than 96 inches; the length of the stroke is to be 4 feet 3 inches, and the number of revolutions on trial about 95 per minute, when developing 12,000 I.H.P. The barrels, covers, and ends of all the cylinders are to be jacketed, with steam of boiler pressure for the high-pressure cylinders, not less than 90 lbs. pressure for the intermediate cylinders, and not

less than 45 lbs. for the low-pressure cylinders. The working barrels of all the cylinders are to be made separately of Whitworth's fluid compressed steel. They are to be flanged and securely bolted to the cylinders with countersunk screws at one end, and fitted with an expansion joint at the other. The width of the steam space of the jackets is not to be less than 1 inch. The branch pipes from the main steam pipes to the jackets of the low-pressure and intermediate cylinders are to be fitted with reducing valves of approved pattern, and pressure gauges. The steam ports are to be stayed with iron or steel screwed stays where necessary, and all other parts are to be properly stayed. The cylinder faces are to be made, separately, of hard close-grained cast iron, and secured to the cylinders with a sufficient number of gun-metal or naval brass screws with recessed heads.

3. *Cylinder fittings.*—The high-pressure and intermediate cylinders are to be fitted at each end with two escape valves, and the low-pressure cylinders with one at each end. These valves are to be provided with springs long enough to permit of the full opening required for the escape of water without undue increase of load on the valves, and to have metal cases and pipes to prevent persons being scalded by any escape of hot water. Suitable apparatus is to be provided for taking indicator diagrams from each end of the cylinders. Cocks also, with pipes for draining water from the lower ends of the cylinders, their slide casings and steam jackets, and the receivers or pipes between the cylinders, are to be fitted. The pipes from the low-pressure cylinders, the low-pressure slide casings, and the receiver pipes between the intermediate and the low-pressure cylinders are to be led into the condensers and bilge, all the others into the feed-tanks and bilge. All drains are to be fitted so that they may be opened independently, and are to have non-return valves fitted in the

pipes leading to the condensers and feed-tanks. The valves on the cylinder jacket drains are to be conveniently situated to enable the flow of condensed steam to the feed-tanks to be easily regulated. The cylinder drain cocks are to be fitted to be worked from the starting platforms. Man-holes are to be provided in the cylinders and their covers for getting at each side of the pistons. The receiver pipes between the cylinders are to be fitted with compound gauges, and safety valves loaded to 90 lbs. per square inch for those between the high and intermediate cylinders, and 45 lbs. for those between the intermediate and low-pressure cylinders. A sight-feed or other suitable steam lubricator, as may be directed, is to be fitted to each high-pressure cylinder. Chocks are to be fitted to the cylinders and any other parts of the machinery where necessary to stay them when the ship is used for ramming.

4. *Holding-down bolts.*—All holding-down bolts for securing the engines in the ship are to be fitted with lock-nuts or guards, to prevent the possibility of their becoming loose when the engines are at work.

5. *Slide-valves and gear.*—The slide-valves are to be double-ported, made of hard close-grained cast iron, planed to a true surface, and fitted with packing rings on the back to relieve them of part of the steam pressure. Other valves of approved construction, in lieu of ordinary slide-valves, may be fitted, if desired. The valve casings are to be so arranged that the valves may be conveniently removed, replaced, and adjusted. The slide-valves are to be fitted with efficient means of lubrication, both when the engines are at work and when cold. Double eccentrics are to be used for working the slide-valves; apparatus is to be provided for fixing the links in any position, and the working parts of the link motion are to be made adjustable; the links and valve-rods are to be of wrought steel. The working surfaces are to be made as large as practicable.

Means are to be provided for balancing the weight of the slide-valves and link motion attached. Approved means are to be provided for linking up each slide-valve independently, to equalise as much as possible the power developed in the cylinders.

6. *Starting-gear.*—Double-cylinder steam-engines as well as hand-gear are to be fitted for working the slides, and the reversing arms are to be capable of continuous circular motion. Auxiliary starting-valves are to be fitted to all the cylinders to facilitate the handling of the engines. The handles for these valves are to be arranged so as to work with the corresponding cranks, and to be placed in the same consecutive order as their corresponding cylinders. Arrangements are to be made for admitting steam direct to the receiver pipes if necessary.

7. *Hot surfaces.*—The cylinders and their covers, the slide-valve casings and covers, and all hot surfaces are to be completely covered with approved non-conducting material and mahogany, properly secured with polished brass strips. The lagging is to be made in portable sections where necessary.

8. *Framing.*—Steel is to be employed, as far as practicable, in the construction of the engine framework and standards to economise weight.

9. *Pistons and rods.*—The pistons are to be of approved construction, made of steel and fitted with single packing rings held out by properly tempered steel springs. The junk-ring bolts are to have secure guards. The piston rods are to be of wrought steel. The rubbing surfaces of the piston-rod crosshead of each cylinder are to be as large as possible, and the guides fitted so as to be capable of adjustment, and to be efficiently lubricated.

10. *Stuffing-boxes and glands.*—All stuffing-boxes are to be made as deep as practicable. The stuffing-boxes of the piston and other large rods are to be fitted with

gun-metal coned rings at their bottoms, and with double glands. The gland nuts of all the principal rods throughout the engines and also those of the stern-tube glands are to be fitted with worm or toothed gearing to enable the nuts of each gland to be screwed up equally, and the rods for working the gear are to be conveniently situated. The glands of the large rods are to have at least four studs and nuts where practicable. The studs of glands not fitted with worm or toothed gear are to have split pins in their ends.

11. *Connecting-rods.*—The connecting-rods are to be of wrought steel. All the steel forgings for the engines are to be made from solid ingots. The length between the centres is to be not less than 8 feet 6 inches. The bearings are to have wrought steel caps, and the crank ends of the connecting-rods are to be made as wide as practicable to give effective support to the brasses. The bearings are to be lined with white metal at the crank-ends.

12. *Crank-shafts.*—The crank-shafts are to be hollow and made of wrought steel. Each shaft is to be made in three parts, which are to be interchangeable. The diameter is to be not less than 16 inches outside, and about 8 inches inside; the aggregate length of the bearings in each set of engines is to be not less than 8 feet 6 inches. The diameter of the crank-pins is to be not less than $16\frac{1}{4}$ inches, and their length not less than 18 inches. The crank-shaft brasses are to be lined with white metal, and so fitted that they may be taken out without necessitating the removal of the shaft. Each top cap is to have a hole in it large enough to admit a man's hand. The crank-pins are to be fitted with centrifugal lubricators.

13. *Propeller-shafting.*—The propeller-shafting is to be hollow and made of wrought steel, and to have solid couplings where practicable and steel bolts. The diameter of the shafting is to be not less than 15 inches outside, and

about 8 inches inside, but the diameter of the stern-shafts abaft the stern tube stuffing-boxes is to be not less than 16 inches outside, exclusive of the casing, and about 9 inches inside. Each thrust-shaft is to have collars, giving a thrust surface of not less than 1,400 square inches for each set of engines. The thrust-blocks are to be fitted with movable collars, so as to admit of ready adjustment. The plummer-blocks are to be of cast iron or steel, and lined with white metal. A bearing is to be fitted as close to each thrust-block as possible, so that the thrust-collars may carry no weight.

14. *Turning and disconnecting gear.*—Apparatus for turning the engines by hand, and for disconnecting and holding the propellers, is to be provided, as well as auxiliary engines for turning the main engines, in addition to the usual hand-turning gear. The turning-engines are to be reversible and of sufficient power to turn either set of engines completely round in ten minutes with steam of 60 lbs. pressure. The turning-wheels are to be of cast-steel, and the worms of wrought steel turned. The disconnecting apparatus is to be of simple construction, and the bolts capable of being quickly removed and replaced. A collar-bearing is to be provided abaft each disconnecting coupling.

15. *Stern tubes.*—The tubes through the stern of the ship and through the outer brackets are to be of gun-metal, not less than 1¼ inches thick at the bearings, and the driving shafts within them are to be effectively cased with gun-metal to prevent access of water to the shafts. The casing is to be not less than 1⅛ inches thick at the bearings. Each tube is to be in one length, and if the casings of the shafts are in more than one length the several pieces are to be tightly filleted into each other so as to prevent leakage at the joints. The brackets for supporting the after-ends of the shafts are to be of wrought-iron or cast-steel, firmly secured to the ship. A pipe and

cock are to be fitted to each stern-tube on the forward side of the stuffing-box bulkhead for drawing water from them. Lignum-vitæ is to be used in both ends of the stern-tubes and in the tubes through the outer brackets; the length of bearing in the brackets is to be not less than 4 feet, at the after end of the stern-tube 3 feet, and at the forward end 2 feet.

16. *Screw-propellers.*—The screw-propellers are to be four-bladed, about 16 feet in diameter, made of gun-metal, and arranged to work outwards when driving the ship ahead. The blades are to be attached to the bosses by screw-bolts, secured by proper guard-plates and screws, and arranged to enable the pitch to be varied as may be required.

17. *Main condensers.*—The main engine condensers are to be made of brass containing not less than 70 per cent. of copper in its composition, and are to be so arranged that the tubes may be readily packed at each end, and taken out without removing any part of the engines which cannot be removed and replaced with facility. Holes are to be provided to admit of the interior of the condensers and the condition of the tubes being examined, and of the deposit being cleaned out from all parts. Cocks are to be fitted for the purpose of readily clearing each condenser of the circulating water when the engines are stopped. Suitable pipe-connexions, with shut-off valves, are to be made between the main steam-pipes in the engine-rooms and each main condenser, to be used as *silent blow-off* when the engines are suddenly eased or stopped. Suitable fittings are to be provided for the purpose of admitting an alkaline solution into the exhaust passages. No iron is to be used in the internal fittings of the condensers. A compound gauge, as well as a vacuum gauge, is to be fitted to each condenser. Directing plates are to be fitted to cause the circulating water to pass over the whole of the tube surfaces. A small valve and pipe are to be

fitted to each condenser to admit water from the circulating chamber to the condensing space or hot-well, to make up loss from leakages and other causes. The tubes are to be of solid-drawn brass, untinned, 0·050 inch in thickness, and to have at least 70 per cent. of best selected copper in their composition. Samples of at least 4 lbs. weight will be subjected to test and analysis. The tube-plates are to be of brass, and supporting diaphragms of brass are to be fitted if necessary. The tubes are to be secured to the plates by tape packings and screwed glands, and approved means are to be adopted to prevent the tubes slipping through the tube-plates. The aggregate area of the cooling surface of the main condensers is to be about 15,000 square feet. The air-pumps, air-pump plungers, and the seats and guards of the foot and discharge valves are to be of gun-metal, and the air-pump rods of gun-metal or rolled naval brass. The plungers are to be provided with efficient packing. If preferred, metallic valves of approved description may be used for the air-pumps instead of india-rubber valves.

18. *Auxiliary condenser.*—An additional condenser, as may be approved, having about 800 square feet of cooling surface, constructed of brass of the same composition as the main condensers, is to be placed in one engine-room for the purpose of condensing the steam used in all the auxiliary engines throughout the ship, and fitted with a separate circulating engine and pump, a suitable pump for delivering the condensed steam into the feed-tanks, and a vacuum gauge.

19. *India-rubber valves.*—All vulcanised india-rubber valves are to be made of the best Para caoutchouc, and with no other ingredients whatever than sulphur and white oxide of zinc; the sulphur not to exceed 3 per cent., and the oxide of zinc 70 per cent. The vulcanised india-rubber is not to contain any re-manufactured materials, but is to

be made of the best ingredients, to be of a homogeneous character throughout, thoroughly compressed, free from air-holes, pores, and all other imperfections, and must be capable of enduring a dry heat test of 270° Fahr. for one hour, and a moist heat of 320° Fahr. for three hours, without impairing its quality.

20. *Circulating engines and pumps.*—The centrifugal pumps for circulating water through the condensers, with their outer cases and fittings, are to be made of gun-metal and to be driven by independent engines. The circulating engines and pumps are to be fitted in duplicate for each set of engines; i.e. there are to be two engines and two pumps for each engine-room, each large enough for full-power working. The engines are to be supplied with steam from branches on the main steam-pipe, they are to exhaust into the condensers as well as into the waste steam-pipe, and are to be placed at as high a level as possible in the engine-rooms. A pipe with screw-down non-return valve is to be fitted to each main circulating pump to draw water from the bilge, in the event of a leak in the ship. The diameter of this pipe is to be not less than the sea suction to the pump, and non-return valves are to be fitted at the lower ends of the bilge suction pipes, to prevent water passing into the ship when the sea suction is open. The bilge suction valves are to be placed in easily accessible places, with their handles above the floor-plates. The ends of the suction pipes are to be placed in convenient positions, and covered with zinced iron or steel strainers. Each circulating pump engine is to be made sufficiently powerful to discharge 1,000 tons of water per hour with steam of 90 lbs. pressure per square inch: trials will be made to test the efficiency of the pumps and engines in this respect. The speed of the engines on these trials is not to exceed 250 revolutions per minute. Approved arrangements are to be made to admit of the discharge of bilge water directly overboard instead

of passing it through the condensers, and means are to be provided for readily shutting off the pumps from the sea and connecting them with the bilge.

21. *Main feed-engines.*—A double-cylinder pumping engine is to be fitted in each boiler-room for feeding the boilers, and is to draw from the feed-tanks and from the bottoms of the condensers. Each feed-pump delivery pipe is to be fitted with a pressure gauge graduated to twice the boiler pressure. A suitable air-vessel and a screw-down non-return valve are to be fitted close to the pump in each delivery pipe. The suction pipes of these pumps are to be fitted with stop-valves, and separate delivery pipes with non-return stop-valves are to be fitted for each boiler-room. The feed engines are to be capable of feeding each or any of the boilers.

22. *Feed-tanks.*—A feed-tank of wrought iron or steel, not less than $\frac{5}{16}$ inch in thickness, and about 100 cubic feet capacity, is to be supplied and fitted in each engine-room as a reservoir for the feed-water from the condensers. Blocks of rolled zinc 12 inches by 6 inches by $\frac{1}{2}$ inch thick are to be suspended in each tank to prevent corrosion. A connecting pipe with a stop-cock in each engine-room is to be fitted between the feed-tanks. Each tank is to be fitted with a stop-cock in the supply pipe, a suitable overflow pipe, and a glass water-gauge.

23. *Auxiliary feed-engines.*—A double-cylinder auxiliary engine is also to be fitted in each boiler compartment, and connected with a set of feed-pipes, feed and overflow valves, separate and distinct from the pipes and apparatus belonging to the main feed-pumps. These pumps are to be fitted to draw water from the sea, and also from the feed-tanks. Each pump is to be fitted to supply any of the boilers; a shut-off valve is to be fitted in the auxiliary feed-pipe in each boiler-room.

24. *Bilge engines.*—Four additional auxiliary engines

(two in each engine-room) are to be supplied and fitted as auxiliary fire-engines for washing decks and pumping out the bilge and boilers. These pumps are to have separate suction and discharge pipes; the discharge pipes to the deck are to be connected with the fire-main. The suction pipes are to be arranged so that the pumps may be used for pumping the water off the inner bottom, and also out of the main drain pipe and the screw tunnels. A pressure gauge and suitable air-vessel are to be fitted to the delivery pipe of each pump.

25. *Fire-engines and fire-main.*—Two steam pumping engines, one in each engine-room, are to be supplied and fitted as fire-engines, and for pumping out the ship and boilers, washing decks, &c. Each of these pumps is to have separate suctions leading to the sea, bilge, and main drain pipe, with suitable stop-valves in each, and to be fitted to discharge into the fire-main, and also direct overboard as may be required. A pressure gauge and suitable air-vessel are to be fitted to each delivery pipe close to the pump. The fire-main is to be 5 inches in diameter, placed as far below the protective deck as practicable, and to extend throughout the ship in the manner which is usual in this class of vessel. It is to be connected with both fire-engine pumps and with all the other bilge pumping engines of the ship, except the circulating pumps for the condensers. Stop and non-return valves are to be fitted in the delivery pipes from each pump leading into the fire-main. T-shaped pipes are to be supplied for the nozzles on the upper deck for receiving the hoses. Delivery pipes of the same size as the fire-main are to be fitted, leading direct overboard, to be used in the event of the pumps being required to pump water out of the ship. Stop and non-return valves are to be fitted in the fire-service pipes where required, and also in the fire-mains running fore and aft. A pressure gauge, graduated to 200 lbs. per square

inch, is to be fitted on the fire-main in each engine-room. All the pipes in connection with the fire-service are to be of copper, and the valves and nozzles of gun-metal. Leather-faced bib-valves, about 1 inch diameter, are to be attached to the fire-service pipes in the principal compartments, to enable water to be drawn off into the fire-buckets.

26. *Hand pumps.*—A hand pump, 6 inches diameter and 6 inches stroke, is to be fitted in each engine-room and connected to the same feed-pipes as the main feed-pumps. These pumps are to be arranged to pump water into the engine-room, or overboard, through independent discharge pipes, and to draw water from the boilers, from the bilge and screw tunnels, or from the sea. Each pump is to be adapted, by means of an additional plunger, 3 inches in diameter, for proving the boilers by water pressure to double the working pressure. Branches with nozzles, caps, and stop-valves are to be fitted in the engine-rooms to the hand pump and the bilge and fire-engine discharge pipes, to admit of hoses being attached. These branches, the nozzles on the deck, and all other nozzles are to be of standard sizes and patterns.

27. *Escape and non-return valves.*—The steam pumps are to be fitted with escape valves where necessary. Non-return valves are to be fitted in the bilge suction pipes, to guard against the danger arising from leaving open the sea-cocks. All escape valves are to be placed in positions easy of access, and so constructed that the valves and springs may be adjusted from the outside. Pipes are to be fitted to the escape-valve boxes of the main feed-pumps to return the escaped water to the suction chambers. All escape-valve boxes, valves, and their casings are to be of gun-metal.

28. *Pipes, &c.*—The steam and all other pipes, as well as their branches, are to be of copper or gun-metal, except where otherwise specified. All feed, bilge, blow-off, and

brine pipes, not exceeding 5 inches diameter, are to be of *solid-drawn* copper. All the nozzles which pass through the ship's side or bottom, and the sea-suction gratings, are to be of gun-metal. The lower end of each bilge suction pipe is to be of zinced or enamelled wrought iron or steel, or other approved material, and suitable strainers of the same material are to be fitted to prevent choking. Care is to be taken that no copper or metal pipes in the bilge rest in contact with the hull; all such pipes exposed to the action of the bilge water should be well painted or varnished, and then covered with waterproof canvas. The flanges of all steam and other pipes are to be faced and the joints made with red lead or wire gauze and not with canvas or millboard; those under the floor-plates are to be fastened with gun-metal or naval brass bolts and nuts. The aftermost lengths of the steam-pipes are to be fitted with ring pieces between the flanges to admit of blank flanges being inserted for testing the boilers and pipes independently of the engines; suitable blank flanges are to be supplied. All steam and eduction pipes and the hot parts of the condensers are to be thoroughly clothed with some approved non-conducting material.

29. *Auxiliary steam service.*—A set of steam pipes, about 7 inches in diameter, is to be provided and connected to the main steam-pipe in each engine-room by a stop-valve. These pipes are to supply steam to the fan, fire, bilge, steering, capstan, hydraulic, air-compressing, ventilating, and all other auxiliary engines throughout the ship (except the circulating and main feed engines, which are to be supplied direct from the main steam-pipe), also to the ejectors, steam whistle and siren, etc., and they are to be fitted with reducing valves where required, and with stop-valves on the branches in all positions considered necessary. A screw-down valve is to be fitted as near as possible to the steam and exhaust orifice of each of the auxiliary engines

throughout the ship, to enable the engine to be shut off from the exhaust as well as from the steam-pipe when not at work. The exhaust steam from all auxiliary engines fitted throughout the ship is to be delivered into the auxiliary condenser as well as into the auxiliary waste steam-pipe.

30. *Steam traps and drain pipes.*—All pipes which convey exhaust steam from the auxiliary engines to the waste steam-pipe are to be fitted with drain valves. Steam traps are to be fitted to the steam-pipes of the auxiliary engines to keep them free from water. All these drains are to deliver into a main drain pipe about 3 inches diameter, running fore and aft, and a zinced iron or steel drain tank of about 40 cubic feet capacity is to be fitted in a suitable position to receive the condensed water. The drains from the safety-valve boxes, whistle pipe, &c. are also to deliver into this main drain pipe. One of the pumping engines is to be connected to this tank to enable the water to be pumped back to the boilers.

31. *Expansion joints.*—Expansion joints are to be fitted to the waste-water pipes, steam and eduction pipes, and to any other pipes where they may be considered necessary; and approved means are to be adopted to prevent fracture by the contraction or expansion of any other parts of the machinery, or by the working of the ship. Tie rods and flanges are to be fitted to expansion joints where necessary.

32. *Regulating valves.*—In the steam pipe leading to each high-pressure cylinder, and as close to the valve-chest as possible, a double-seated equilibrium valve is to be fitted. These valves are to have connections leading to the platforms to admit of their being used to regulate the supply of steam to the engines. The valves and valve-chests are to be made of gun-metal.

33. *Separators.*—A copper separator is to be fitted to each main steam-pipe. The separators are to be arranged to discharge into the sea, with an additional branch and

stop-cock with non-return valve leading to the feed-tanks. The sea discharge pipes are to be attached to a separate sea valve or the nearest discharge valve, as may be determined, and to be fitted with stop and non-return valves. A glass water-gauge and a small drain cock are also to be fitted to each separator.

34. *Underwater valves.*—Efficient stop-valves are to be fitted at the ship's side to the circulating-pump discharge pipes; the valve boxes are to be made of gun-metal, and non-return valves are to be fitted to the extremities of all pipes discharging into them. Kingston's valves, with the spindles cast on, or screw-down valves, furnished with stop-cocks or valves, are to be fitted close to the skin of the ship, to the blow-off pipes, and to the suction-pipes of the circulating pumps, auxiliary pumps, and hand pumps, and, if required, to any other pipes. All underwater valve spindles are to be proved by a tensile strain of half a ton for every square inch of area of the valve, but in no case need the test exceed twelve tons. No waste water is to be delivered above the armour shelf. All valves and cocks in connection with the boilers or ship are to have spigots passing through the plates. Gun-metal or naval brass bolts only are to be used in connection with all gun-metal underwater fittings of the ship. Zinc protectors are to be fitted at all the openings in the ship's skin for the underwater valves. The handles to all cocks and valves in the engine department which open communication with the sea and bilge are to be placed at a convenient height above the platform, or flooring plates in their vicinity, in positions easily accessible and above the level of the fire-bars. They are to be so fitted that the valves will be closed by a right-hand motion, and suitable means are to be fitted for quickly changing the suctions from the sea to bilge when necessary. Brass index plates are to be fitted to all the principal cocks and valves in the engine and boiler rooms.

35. *Governors.*—A governor, on an approved plan, is to be fitted for each set of engines, together with a properly balanced valve and all the necessary connections. The governors are to be of sufficient power to prevent the engines from racing in stormy weather.

36. *Telegraphs and voice tubes.*—An engine-room telegraph of approved pattern, with reply gongs and any number of dials that may be found necessary, is to be supplied and fitted for each set of engines. Suitable gear is to be provided, to enable any of the dials to be disconnected from the engine-room when not required. Telegraphs of approved pattern are also to be fitted from the engine-rooms to the stokeholds. A sufficient number of voice-pipes of copper, not less than 0·036 inch in thickness, are to be provided and fitted in a suitable manner, to put the engine and boiler rooms in communication, and also to communicate from the bridge, the conning tower, and, if ordered, from any other part of the ship to the engine-rooms. Each voice-pipe is to be fitted with mouthpieces and whistles, or otherwise as required. These pipes are to be cased where necessary.

37. *Counters and tell-tales.*—A counter, on an approved plan, capable of registering up to one million, is to be supplied for each set of engines; also a tell-tale for each set to show the direction of motion of the engines, with an index on the bridge, and an index in the conning tower. Revolution or speed indicators on an approved plan, with any number of dials that may be required, are to be fitted for each set of engines.

38. *Savealls.*—Copper or brass savealls are to be fitted for all the working parts of the engines, and for the bearings of the screw shafting; the oil from the central parts of the engines to run into one large saveall in each engine-room, dished in the lower part. A bilge engine is to be connected to each dish, so that the contents

may be pumped out while the engines are at work. Save-alls for oil cans and for trimming lamps are to be fitted as required.

39. *Plugs for discharge pipes.*—A gun-metal plug with an india-rubber joint ring is to be fitted for each size main discharge pipe, to be put in or taken out by means of a lever on the ship's side above the water-line. The out-riggers on the ship's side are to be fitted complete with all levers and gear.

40. *Water pressure tests.*—The boilers, steam-pipes, and all connections up to the cylinders are to be tested by water pressure to 260 lbs. per square inch. The high-pressure cylinders, covers, valve-casings, and all auxiliary engines, steam jackets, pipes, and connections, valve boxes, impermeators, &c., subjected to the pressure of the boilers, are to be tested by water pressure to 230 lbs. per square inch; the intermediate cylinders, covers, jackets, and the receivers or pipes between the high-pressure and intermediate cylinders, to 180 lbs. per square inch; the low-pressure cylinders, covers, jackets, and the receivers or pipes between the intermediate and low-pressure cylinders, to 130 lbs. per square inch; the condensers to 30 lbs., the condenser tube casings and feed-tanks to 15 lbs. per square inch. The cylinders and condensers are also to be tested by water to the required pressures before being put on board, and are to be so placed that all parts may be accessible for examination during the test. The pumps, valve-boxes, and air-vessels of the fire and bilge engines are to be tested by water pressure to 260 lbs. per square inch. Each of the boiler and condenser tubes is to be tested by water pressure separately to 300 lbs. per square inch. No lagging or covering of any kind is to be in place on the cylinders, pipes, or condensers during the water test.

41. *Boilers.*—The boilers are to be eight in number, of cylindrical type, about 15 feet 2 inches in diameter, any

one of which may be used independently of the others. They are to be capable of carrying steam of 130 pounds working pressure per square inch, and are to be proved by water pressure to 260 pounds per square inch, and are not to be painted until they have been satisfactorily tested. After the boilers have been proved, they are to be well painted with red lead or oxide of iron paint; they are afterwards to be clothed in an approved manner on the barrels and ends with incombustible non-conducting material, secured by steel angles, and covered if considered necessary with zinced sheet iron or steel, about $\frac{1}{16}$th of an inch thick. No combustible material is to be fitted in connection with the clothing of the boilers or steam-pipes. The fronts of the boilers are also to be clothed with approved non-conducting material and iron or steel where considered necessary. Steam is not to be raised in the boilers until after the water pressure tests have been made, unless desired for drying joints, for which purpose the pressure must not exceed 10 pounds per square inch. The boilers are to be capable of supplying steam for all the auxiliary engines, as well as the main propelling engines when the latter are working at full power. There are to be thirty-two corrugated steel furnaces, each about 3 feet 2 inches in diameter and 7 feet long. The furnace frames and doors are to be made of wrought-iron or steel, and fitted to enable the doors to be kept open in a seaway, and arrangements are to be made for the admission of air through the frames and doors above the fires. Each boiler is to be fitted with a portable damper in the uptake, at least two safety-valves, and two pressure-gauges, one of which is to be graduated to 160 pounds and the other to 260 pounds, and provided with 3-way cocks and flanges for application of the test gauges. Two sheet-brass internal steam-pipes, with transverse slits on their upper sides only, through which the steam must pass, are to be fitted to

each boiler, and are to extend over its whole length. A damper is to be fitted on the front of each ash-pit, and water-troughs for cooling ashes as may be directed. Internal plates are to be fixed to the smoke-box doors. Glass water-gauges (two to each boiler), gauge and test-cocks, and internal brass feed pipes, are to be provided and fitted. A sufficient number of slabs of rolled zinc 12 inches by 6 inches by $\frac{1}{2}$ inch thick is to be suspended in each boiler, in metallic contact with the plates and stays, for the purpose of preventing corrosion, and thin iron plates are also to be fixed in proper positions to assist the circulation of the water. The area of the tube-surface, between the tube-plates, is to be not less than 17,000 square feet, and the area of the fire-grate is to be about 690 square feet.

42. *Boiler plates.*—The boilers are to be constructed entirely of steel made by the Siemens-Martin process. The minimum thicknesses of the boiler plates are to be as follows:—Shells $1\frac{3}{16}$-inches, tube plates and ends $\frac{7}{8}$-inches, furnaces $\frac{1}{2}$-inch, and all other internal parts $\frac{1}{2}$-inch. The longitudinal seams of the shell plates are to be butt-jointed with double butt-straps, and treble-riveted, the other seams are to be double-riveted. The furnace bars are to be of wrought iron $3\frac{1}{2}$ inches deep by 1 inch wide, and made in three lengths.

43. *Tests for plates, stays, &c.*—Strips cut lengthwise or crosswise are to have an ultimate tensile strength of not less than 26 tons, and not more than 30 tons per square inch of section, with an elongation of 20 per cent. in a length of 8 inches. The angle, tee, and bar steel and rivets are to stand such forge tests, both hot and cold, as may be sufficient to prove soundness of material and fitness for the service intended. Strips cut lengthwise or crosswise, $1\frac{1}{2}$ inches wide, heated uniformly to a low cherry red, and cooled in water of 82° Fahrenheit, must stand bending double in a press to a curve of which the inner radius is

one-and-a-half times the thickness of the steel tested. The strips are all to be cut in a planing machine, and to have the sharp edges taken off. In addition to these quantitative tests the ductility of every plate, angle, &c. is to be ascertained by the application of one or both of these tests to the shearings, or by bending them cold by the hammer. The pieces of plate, angle, &c. cut out for testing are to be of parallel width for 8 inches of length. All steel is to be free from lamination and injurious surface defects. The furnaces and combustion-chambers are to be made of steel of 'special soft' quality. In addition to the above tests these plates are to be tested by welding and forging, and some of the welded pieces are to be broken in the testing machine to ascertain the efficiency of the welding. The tensile strength of these plates should not exceed 25 tons per square inch. All steel castings for engines are to satisfy the following conditions:—Tensile strength not less than 28 tons per square inch, with an extension in 8 inches of length of at least 10 per cent. Bars of the same metal $1\frac{1}{4}$ inches square should be capable of bending cold without fracture, through an angle of 90° over a radius not greater than $2\frac{1}{2}$ inches. Test-pieces will be taken from each important casting.

44. *Rivet-holes.*—The holes in the plates, angles, &c. are to be drilled and not punched, and are to be drilled in place after bending. In parts where the rivets are not closed by hydraulic riveting machinery the rivet-holes are to be countersunk, and coned rivets used.

45. *Boiler tubes.*—The tubes are to be of iron or steel of approved brand. They are to be not less than $2\frac{1}{2}$ inches in diameter, 0·165 inch mean thickness (4 lbs. weight per foot run), and not less than 1 inch apart. The smoke-box ends of the tubes are to be $\frac{1}{16}$ inch larger in diameter than the body of the tube. The stay tubes are to be not less than $\frac{1}{4}$ inch in thickness (5·9 lbs. weight per foot run),

and they are to be so fitted in the tube-plates that they may be removed and replaced with facility. The ends of the stay tubes are to be thickened for screwing, so that the thickness at the bottom of the thread is not less than ¼ inch. The distance between the several rows of tubes in the boilers should not be less than 1 inch.

46. *Man-holes and stays.*—In the manufacture of the boilers sufficient room is to be provided for man-holes at the upper parts and the lower parts of the furnaces, both for the purposes of cleaning and repairing. All man-holes and mud-holes at the ends of the boilers are to have outside stiffening rings. The doors are to be of wrought steel, and placed on the inside of the boilers. The man-hole frames on the tops of the boilers are to be raised about 7 inches, and the covers secured with bolts and nuts. The stays are to be arranged on an approved plan, to admit of easy access to the internal parts of the boilers. The long stays are to be secured to the shells of the boilers by tee steels or in other approved manner; the rivets or bolts for securing the stays are to be at least 25 per cent. stronger than the stays. Palm stays, if fitted, are to be forged from the solid, and not welded. All short stays are to be screwed to a pitch of eight threads per inch. The screwed stays are to be nutted on all flat surfaces. The holes for the screwed stays in the water spaces are to be drilled and tapped together after the furnaces and combustion-chambers have been riveted in place in the boilers. The maximum stress per square inch at the smallest section of the steel stays at the working pressure must not exceed 9,000 lbs. for stays above 1½ inches diameter, and 8,000 lbs. for stays not exceeding 1½ inches diameter.

47. *Blow-out apparatus.*—Surface blow-out, as well as bottom blow-out, apparatus is to be fitted to each of the boilers, and the pipes are to be so arranged that each boiler may be blown out separately. The blow-out sea-cocks are to

be fitted with guards over the plugs with featherways, the spanners having corresponding feathers so that they cannot be removed unless the cocks are shut. Screw-down valves are to be fitted on each boiler shell for the surface and bottom blow-outs.

48. *Shut-off valves and steam-pipes.*—The main steam-pipes are to be connected athwartships as required, and self-closing shut-off valves fitted in them so that, in the event of any one of the boiler-rooms becoming flooded, or its steam pipe damaged, its steam connection may be shut off, and the steam from the remaining available boilers conveyed independently either to the port or to the starboard engines. Additional screw-down valves are to be fitted between these self-closing valves and the regulating valves at the engines. The steam-pipes in the boiler-rooms are to be well clothed with some approved non-conducting material; those passing through the after boiler-rooms are to be covered in addition with some waterproof material. The flanges are to be clothed separately. The shut-off valves are to be fitted to be worked from the lower deck and engine-rooms.

49. *Safety-valves, springs, &c.*—The safety-valves are to be loaded with springs on an approved plan, and placed on the boilers independently of the internal steam-pipes. The area of the safety-valves on each boiler is not to be less than 30 square inches. A pipe connected to each safety-valve box is to be provided for conveying the condensed steam to the tank mentioned in par. 30. Gear is to be fitted to enable the safety-valves to be relieved from the boiler-room floors and lower deck; the gear from each place is to work independently. All the joints of the safety-valve easing gear are to be bushed with gun-metal or fitted with gun-metal pins. Ferrules suitable for the reduction of pressure (by decrements of 5 lbs.) to 90 lbs. per square inch are to be supplied for all the valves. The

springs are to be made so that the amount of compression when they are screwed down to the maximum working load shall not be less than the diameter of the valves. Provision is also to be made to prevent the valves being blown out in the event of the springs breaking.

50. *Feed valves, &c.*—The main and auxiliary feed valves are to be constructed so that they may be regulated by screws. The main feed valves are to be placed on the right-hand sides of the boilers, and the auxiliary feed valves on the left-hand sides. Index plates are to be fitted on the fronts of the boilers at each gauge glass to indicate the position of the top of the combustion-chamber or the highest part of the heating surface. An air-cock is also to be fitted on the highest part of each boiler shell. Valves with fittings complete are to be provided for cleaning the tubes by a steam jet. A fog whistle, 9 inches in diameter, and a steam siren for signalling purposes, are to be supplied and fitted with steam valves and pipes complete. Gear is to be fitted to enable the whistle to be blown from the bridge. A small cock is to be fitted in the front of each boiler to enable water to be drawn off for testing. Small cocks are to be provided for effectually draining the steam-pipes, stop-valve boxes, and whistle pipe.

51. *Boiler mountings.*—All feed, blow-off, stop, and safety valves, and their boxes, and the boiler mountings generally, are to be made of gun-metal. All bolts which penetrate the shell of the boiler for securing the boiler mountings, lagging, or any other purpose are, where practicable, to be screwed through the plate with the bolt-heads inside. All valve boxes and covers are to be secured by outside flanges where practicable, or if otherwise attached, care is to be taken that the bolts do not penetrate into steam or water spaces. Any internal apparatus for working the safety valves is to be of gun-metal, and all pipes inside the boilers are to be of brass.

All valves and cocks in connection with the boilers are to have spigots passing through the plates.

52. *Stop-valves, self-closing.*— The stop-valves on the boilers are to be made self-closing, of approved construction, and placed with the spindles horizontal. The main valves are to have suitable index plates, and fitted so that they may be closed from the debris deck.

53. *Funnels and casings.*—The funnels are to be fixed, and exterior casings are to be fitted round them to carry off the hot air from the boiler-rooms. An armour grating is to be fitted in the funnels and air-casings at the level of the armour-deck. All the fittings necessary for securing the funnels are to be fitted complete. The funnels below and above the armour-gratings are to be made of $\frac{3}{16}$-inch steel plate, stiffened as may be required by angles, and the casings of 0·100-inch steel plate. A cover on an approved plan for the top of each funnel is to be supplied and fitted with a suitable drain for carrying away rain-water. The main waste steam-pipes from the safety-valves and the auxiliary waste steam-pipes from the auxiliary engines are to be taken up separately through the casings outside the funnels. The height of the funnels from the dead plates of the lower furnaces is to be about 65 feet. The tops of the casings round the funnels on the upper deck are to be fitted with gratings and movable covers, and openings (fitted with sliding doors) are to be made in them, at the sides and after-ends.

54. *Coal-boxes.*—The coal-boxes are to be watertight, and fitted with suitable sliding-doors, and with air-tubes for ascertaining their temperature; the tops of the tubes are to be fitted with screwed deck plates. Suitable ventilating pipes are also to be fitted to allow the gases evolved from the coals to pass away to the atmosphere; the ventilating pipes from the several bunkers are to be independent of each other. Rails, trucks, and the necessary

fittings are to be provided and fixed in the bunkers; and suitable arrangements are to be fitted, with the necessary carriages and buckets, for carrying coals to the furnaces. The boiler-room floor-plates are to be of wrought iron or steel, ribbed, not less than $\frac{3}{8}$ths of an inch in thickness.

55. *Ash hoists.*—Ash trunks, and steam hoists complete to suit them, are to be provided and fitted. The ash trunks are to be fitted so that they may be used when the stokeholds are worked under air pressure.

56. *Bilge ejectors.*—Four ejectors of approved pattern, each capable of discharging 300 tons of water from the bilges per hour, are to be supplied and fitted in place, with all necessary tanks, steam pipes, and other fittings. The diameter of the steam-pipe to each ejector is to be about 5 inches.

57. Pipes with suitable cocks or valves, fitted with hose connections of Admiralty pattern to enable flexible hoses to be attached, are to be supplied and fitted for the purpose of filling the boilers of small steamboats with hot water from the main boilers, and also to permit the water in the boats' boilers to be boiled by the admission of steam, as may be required.

58. *Ventilation.*—Fans, with small engines to drive them, for the ventilation of the engine and boiler rooms, and also for the ventilation of the ship, are to be provided and fitted complete, with all steam and exhaust pipes, and supply trunks and distributing pipes. Each fan engine is to be fitted with a sight-feed or other suitable steam lubricator. All the necessary ventilating tubes, to supply sufficient air to the engine-rooms, boiler-rooms, and furnaces, when not working under air pressure (with deck fittings), the casings and screens in the boiler-rooms, and the casings and cowls above the upper deck, are to be supplied and fitted. The cowl heads of the ventilating tubes are to be carried well above the tops of the bulwarks

of the ship, and fitted with friction rollers and suitable turning gear.

59. *Fittings for forced draught.*— Fans, with small engines to drive them, are to be provided and fitted for supplying forced draught to the boiler-rooms. The fan engines should be capable of maintaining an air pressure in the boiler-rooms equal to a head of water of at least 3 inches. Each fan engine is to be fitted with a sight-feed or other suitable steam lubricator, also with a tachometer or other approved revolution indicator. Suitable dampers are to be fitted, so that each fan may be shut off when necessary. All necessary casings, screens, air-locks, and other fittings required in connection with the boilers for working under air pressure are to be supplied and fitted.

60. *Spare gear.*—Spare gear is to be provided for all the important working parts of the machinery; the several articles to be accurately fitted and tried in place. The whole is to be stowed on board the ship in convenient places, as may be arranged, and firmly secured by straps and bolts, or by any other means which may be considered necessary.

61. *Store tanks.*—Tanks are to be supplied and fitted to hold the quantity of oil, tallow, paint, oakum, india-rubber valves, and other engineers' stores required by the establishment for six months' use. These tanks are to be fixed in the engineers' store-room, in the screw tunnels, or elsewhere, as may be directed, with brackets, straps, and the necessary fastenings for securing them in place. The tanks not placed in the engineers' store-room are to be provided with locks or other means of securing them.

62. *General.*—All requisite lubricators and water service pipes for the engines and shafting, gauges fitted with lamps of Admiralty pattern, and brass covers, vessels for supplying the slide-valves and cylinders with oil, and all necessary appliances for the ready removal of the cylinder

covers, pistons, and other parts of the engines, are to be supplied and fitted. The glasses of the water gauges on the boilers, separators, and feed tanks are to be $\frac{5}{8}$-inch external diameter, and are to have white enamelled strips. The contractors are to supply and fit in place all the ladders, platforms, and floor-plates required for the engine and boiler rooms, the fenders and guard rails for the engine-rooms, boiler-rooms, and shaft passages, the spanner racks in the engine-rooms, with two complete sets of spanners, and the necessary racks for the stowage of the fire-irons in the boiler-rooms. The screw threads and the dimensions of all ordinary bolts, nuts, and pins used in the engines and boilers, and in every other part of the work furnished by the contractors, are to agree with Whitworth standards. All iron bright nuts for screws $\frac{1}{2}$-inch diameter and under are to be case-hardened. Such parts of the engines, boilers, and other work supplied by the contractors as are usually painted are to have at least three coats of paint. The boilers are not to be painted until they have been proved by water pressure.

63. *Distilling condensers.*—Two distilling condensers for distilling fresh water from sea water, for drinking, washing, and culinary purposes, are to be supplied and fitted, one in each engine-room: these condensers are to be fitted with aërating pipes, charcoal filters, and receiving tanks, and each is to be capable of producing at least 150 gallons of pure fresh water per hour, from steam at a pressure of 15 lbs. per square inch. Tests will be made to ascertain the efficiency of the condensers in this respect. Both condensers are to be connected direct to the boilers by separate steam-pipes and the necessary shut-off valves. A connection is to be made between the suction pipe of one of the bilge pumping engines and the outlets of the distilling condensers, to provide for the circulation of cold sea-water through them. This bilge engine and also an

auxiliary feed-engine are to be connected with the distilling steam-pipe as well as with the auxiliary steam-pipe, the distilling service of steam-pipes being made large enough for this purpose. A small engine and pump, with all pipes, &c., complete, for pumping the fresh water from the receiving tanks into the ship's tanks, are to be provided and fitted.

64. *Auxiliary machinery.*—Arrangements are to be made for working the following auxiliary machinery. If possible, central pumping engines are to be fitted to enable the whole of the machinery to be worked by the agency of hydraulic power or compressed air. If this cannot be conveniently arranged, the system of steam and exhaust piping, &c. for the several engines is to be designed in as uniform and simple a manner as may be practicable. The cylinders of all auxiliary engines worked with steam from the main boilers should be of sufficient size to enable the engines to be efficiently worked with steam of reduced pressure, say 90 lbs. per square inch; and suitable reducing valves are to be fitted on the branches leading to the steering and capstan engines, and in any other cases that may be found necessary, to prevent undue straining of the gear when the full pressure is used. A pressure gauge and safety or alarm valve should be fixed between the stop and reducing valves of each of the auxiliary engines that is so fitted.

65. *Steering engine.*—An efficient steering engine, with all necessary connections and gear, is to be supplied and fitted complete. The engine is to be connected, by means of accurately turned and fitted shafting and gear, to a sprocket wheel, or to a grooved winch barrel for steel wire rope, in the after compartment of the ship, so as to reduce the length of chain or wire rope required. The shafting is to be provided with stuffing-boxes on the watertight bulkheads, through which it may pass. The starting or

reversing valve of the engine is to be balanced, and steering wheels, with gear for working this valve, are to be fixed on the bridges, in the conning towers, and at such other parts of the ship as may be considered necessary. Clutches, or other suitable arrangements, are to be fitted to enable any of the steering wheels to be readily put in or out of gear as may be required. Controlling gear is to be fitted to cause the working of the engine in either direction to always tend to replace the reversing valve in its central position, so that the engine may stop immediately after the steersman ceases to work the steering wheel, and hold the rudder in the required position, until the engine is again moved by the operation of the steering wheel. One steering wheel is to be placed close to the engine, and arrangements are to be made to provide communication between this wheel and the deck and conning towers by means of suitable telegraphs and voice-pipes. Tell-tales worked by the rudder-head are to be fitted to show the angle of helm. The several steering wheel boxes are to be fitted with proper stops or guards to regulate the extreme angle to which the tiller may be placed by the engine, and prevent overstraining of the steering gear. The steering engine is to be of sufficient power to enable the rudder to be placed hard over to starboard or port, as required, in 30 seconds, when the ship is steaming at full speed, and all the gear connected therewith is to be strong enough to withstand this action without any undue straining. Efficient disconnecting arrangements are to be fitted, to allow the change from steam to hand gear, or *vice versâ*, to be readily effected.

66. *Capstan engine.*—An engine, with two cylinders and all necessary connections, is to be supplied and fitted for the purpose of working the ship's capstan, by means of either spur or worm gearing. The engine is to be fitted with reversing gear, and the cylinders are to have suffi-

cient capacity to enable a strain of $\frac{2}{3}$rds the proof test to be brought on the chain cable, when the engines are worked with a steam pressure not exceeding 90 lbs. per square inch. A telegraph and voice-pipe are to be fitted from the capstan engine-room to the forecastle.

67. *Electric lights.*—The electric light installation is to consist of three sets of engines and dynamos, two arc search lights, and about 350 incandescence lamps for internal lighting. Also two yard-arm reflectors, each fitted with eight 40-candle incandescence lamps, with all leads, lamps, switches, brackets, globes, guards, safety plugs, mounts, and all electrical fittings, spare gear, and casings, projectors and hand lamps. Each dynamo and engine must be capable of giving an arc search light of not less than 25,000 candles, or 150 incandescence lamps of not less than 20 candles each when driven at a speed not exceeding 400 revolutions per minute.

The engines are to be direct-acting compound, fixed on a strong cast-iron bed-plate, and coupled direct to the dynamos with flexible couplings. All the working parts are to be of steel, with large bearing surfaces. The shaft bearings are to be of gun-metal lined with white metal, and to be capable of running at full speed with the full load on the dynamo without knocking or vibration. The engines are to be designed of suitable dimensions to drive the dynamos at the specified speed, when exhausting into the atmosphere, with a steam pressure in the boilers of 90 pounds per square inch.

The dynamos are to be of the compound wound direct-current type. Each machine, when running at the speed specified, must be capable of giving and sustaining a steady current of not less than 200 ampères, and a pressure of 80 volts. The dynamos are to be capable of maintaining this current in an atmosphere of 80° Fahrenheit without sparking at the brushes or injurious heating of any part of

the machine. The armature is to be well balanced and fitted, and properly secured to a steel shaft, running in long thoroughly lubricated bearings. The commutator sections may be either of copper or bronze, insulated with mica. The sections are to be not less than 1½ inches deep, and to have a working brush surface not less than 5 inches long. The brushes are to be carried on a rocking bar, two on each arm; each brush is to have a separate spring, with a forward thrust, and to stand at an angle of about 50° to the surface of the commutator.

All wire leads are to be made of the best high conductivity copper, and to form a complete wire circuit, no earth being formed by the ship. The leads to the projectors are to be strands of 19 wires, each of ·072 inch or 15 L. S. G. All branches to single lamps are to be single wires of 18 L. S. G. All other leads are to be strands having an aggregate sectional area of not less than one square millimètre for every two 16 C. P. lamps on the circuit. Leads to the yard-arm reflectors are to be not less than one square millimètre for every lamp. All the leads are to be insulated in the best manner; the thickness of the dielectric is to be at least $\frac{3}{32}$-inch, measured from the conductor. All joints and connections in leads are to be well made and soldered, without the use of acid or zinc solution.

All the casings for the wire leads are to be made of teak, sound and free from shakes. The grooves are to be at least ½-inch apart and to be sunk to a depth equal to 1¼ times the external diameter of the wire lead. The grooves are to be painted with white lead, and the wire leads well bedded in fresh putty and fitted flush with putty before the covering board is fixed. The covering board is to be well fitted and secured with brass screws.

The main switch board is to be fixed in the dynamo room and arranged to take all the circuits; each circuit is to have its own return wire, so that any circuit may be tested

without stopping the machine. The board is to be arranged so that each of the machines may be put on any circuit, and no two on the same circuit at the same time. All terminals and connections on the board are to be of good gun-metal, well secured to a strong slate base. Each circuit from the dynamo room should be arranged so that the current may be safely cut off without stopping the machine.

The glow lamps are to be of high resistance, of best quality, and to be 20- or 10- candle-power as directed, except the yard-arm lamps, which are to be 40-candle-power. All lamps are to have strong platinum wire connections to stand rough wear and the concussion of heavy guns, and to be mounted on flexible springs. All lamps must be of equal resistance, suitable for the dynamo current, thoroughly exhausted of air, and to burn without undue heating of the glass. The lamps in the cabins are to be mounted on bronze brackets of approved design. Those in exposed parts of the ship are to be specially mounted with complete watertight shades, and protected by stout wire guards. About 20 lamps in the boiler and engine rooms are to be portable, with 20 feet of flexible wire attached to each.

The yard-arm lamps are to be mounted in a circular concave brass reflector, about 20 inches in diameter, and 6 inches deep from the centre of the inside of the top, made of brass sheets $\frac{1}{16}$-inch thick, formed of one piece without joint turned over a wire ring at the edge $\frac{3}{8}$-inch diameter, with a stout gun-metal eye at the top, and with two small eyes on the rim for the guys. The circuit wire is to enter at the top and to be watertight. The reflector is to be painted with white enamel inside to reflect the light. Each lamp is to have sufficient flexible braided wire to reach from the terminal box on the deck to the fore or main yard-arm on either side of the ship, and arranged so that one or both lamps may be burnt on either side of the ship.

A fusible lead wire is to be fitted to each main and sub-main, with a sectional area not greater than that of the copper joined. The base of the lead plug is to be slate, and to have a watertight metal cover.

Key switches are to be fitted to all branches to the store-rooms and places that only require lights occasionally, and also to the mess deck circuits and the cabin lamps. The switches are to be of simple construction with slate bases.

68. *Torpedo machinery.*—Suitable engines and pumps for the purpose of compressing air for charging and launching Whitehead torpedoes are to be supplied, and fitted with all necessary air-valves, piping, reservoirs, charging columns, &c., complete. The main reservoir to consist of fifty steel tubes, about 2⅜-inch internal, and 3-inch external diameter, and 6 feet long. The several tubes are to have gun-metal caps screwed on their ends, and to be connected together by short pipes, the total capacity of the reservoir being about 11 cubic feet. The engines are to be of sufficient power to enable the reservoir to be filled with air of a pressure of 1,500 lbs. per square inch in about 20 minutes, when worked with steam of 90 lbs. pressure. The air piping is to be made of solid-drawn copper ⅜-inch internal diameter. Charging columns, with proper air and drain valves, are to be fitted at such parts of the deck as may be necessary from the position of the ports for the torpedo firing carriages; one charging column is to be fitted close to the torpedo room. Voice pipes are to be led from the torpedo pumps to each of the charging columns.

69. *Boat lifting-engines.*—Engines are to be fitted to drive suitable winch barrels and gear for the purpose of lowering the torpedo and other boats, carried by the ship, into the water, or lifting them on board. The winch barrels are to be of large diameter and fitted for the use of steel wire ropes. The engines are to be sufficiently

powerful to lift 12 tons at the rate of 10 feet per second when worked with steam of 90 lbs. pressure. The gear is to be arranged so that the barrel for topping the derrick may be worked independently of the lifting gear.

70. *Winches, &c.*—All necessary winches, hoists, presses, &c., and other machines worked by steam, water, or air power, for lifting ammunition and projectiles, or for any other purposes, are to be provided for in the general arrangement of the auxiliary machinery of the ship, and to be fitted in place with all gear and fittings complete.

71. *Composition of gun-metal.*—The gun-metal used for the screw-propellers and other large castings should have in its composition not more than 5 per cent. of zinc, nor less than 8 per cent of tin, the remainder being best selected copper. For the smaller gun-metal fittings the composition should be: Copper 88, tin 10, and zinc 2 parts. Cuttings will be taken from the principal castings for analysis.

72. The whole of the work in connection with the machinery of the ship is to be of the best materials, construction, and workmanship; and its efficiency is to be guaranteed for a period of twelve months' ordinary work from the date on which it is accepted, after thorough examination and trial of the various parts have been satisfactorily made.

PART V.

THE PROPELLER.

CHAPTER XX.

PROPULSION.

Resistance of ships.—IN considering the question of the propulsion of ships, it will be necessary in the first place to explain briefly the general nature of the resistance experienced by ships in their passage through the water, which resistance has to be overcome by the action of the propeller. The most important element in the resistance to a ship's motion is the frictional action of the water itself on the skin of the ship. Water is not a perfect fluid, and when it is disturbed its particles will exercise friction both on each other and also on the surface of any body with which they may come in contact. If a well-formed ship with a clean bottom be towed through the water, the water will open out at the bow, and follow round the sides of the ship in well-defined currents, called stream lines, closing in again under the stern, so that the counter-pressure under the stern thus caused tends to balance the head resistance to the ship's motion. If the run aft be not sufficiently fine to allow the water to close in under the stern properly, an eddying wake would be formed under it, which would

increase the direct or head resistance to the motion of the vessel. The surface or wave-making action constitutes another source of resistance. From the bows of all ships in motion, waves are formed, to a greater or less extent, which in most cases pass away from the ship in divergent directions, without producing a corresponding reaction under the stern; and the energy expended in creating these waves is evidently wasted.

The three elements constituting the total resistance opposed to the ship's motion are, therefore:

1. Frictional resistance, due to the gliding of the particles of water over the rough skin of the ship.

2. Eddy-making resistance, due to a wake at the stern.

3. Surface disturbance or wave-making resistance.

The first of these is by far the most important. In well-formed ships the second is very small and need only be considered in exceptional cases. The third element, due to surface disturbance, may in some cases be of considerable importance. This cause of resistance may be reduced to a minimum by making the lengths of entrance and run of the ship sufficiently great for her required maximum speed; or, to put it in other words, for a certain speed of a ship there are minimum lengths of entrance and run which must be given in order to reduce the loss from surface disturbance to a minimum. This condition is more affected by the length of run than by that of the entrance; and the entrance may be reduced to a certain extent without entailing so great a loss of efficiency as would result from a similar decrease in the length of the run of the ship.

For every ship there is a certain limit of speed, beyond which any addition can only be obtained at the expense of a very rapid increase in the resistance; and this is attributed to the wave-making action. Up to the above limit

the resistances due to the eddy and wave-making elements are comparatively small, and the total resistance is approximately proportional to the frictional resistance, and varies practically as the square of the speed. Beyond this limit the resistance has been found to vary as the cube and still higher powers of the speed.

Frictional resistance varies with the amount of the immersed surface of the ship, on the coefficient of friction of the skin in water, upon the length of the surface and the velocity with which the particles of water glide over it. The length has an important influence. For example, in some experiments made by the late Mr. Froude, it was found that a plane 8 feet long coated with varnish, moving at a speed of 600 feet per minute, experienced a resistance of 0·325 lb. per square foot, whereas the resistance opposed to a similar plane 50 feet long moving at the same speed was only 0·25 lb. per square foot or one-fifth less. For greater lengths, up to 300 or 400 feet, the resistance per square foot was about the same as for the 50-feet length. The skin-resistance of a ship may be regarded as practically the same as that of a plane of the same area as the immersed surface of the ship, with similar ratios of length to depth.

Mr. W. H. White, in his 'Manual of Naval Architecture,' gives the following summary of the principal facts relative to the total resistance offered to the motion of a ship when towed or propelled by sails through the water. The effect of the action of the propeller in increasing the resistance will be pointed out farther on.

1. 'That *frictional resistance*, depending upon the immersed surface of a ship, its degree of roughness, its length, and (about) the square of the speed, is not sensibly affected by the forms and proportions of ships; unless there be some unwonted singularity of form or want of fairness. For *moderate* speeds, this element of resistance

is by far the most important; for *high* speeds it also occupies an important position—from 50 to 70 per cent. of the whole resistance, probably, in a very large number of cases when the bottoms are clean; and a larger percentage when the bottoms become foul.

2. 'That *eddy-making resistance* is usually small, except in special cases, and amounts to some 8 or 10 per cent. of the frictional resistance. A defective form of stern causes largely increased eddy-making.

3. 'That *wave-making resistance* is the element of the total resistance which is most influenced by the forms and proportions of ships. Its ratio to the frictional resistance, as well as its absolute magnitude, depend upon many circumstances; the most important being the forms and lengths of entrance and run, in relation to the intended full speed of the ship. For every ship there is a limit of speed beyond which each small increase in speed is attended by a disproportionate increase in resistance; and this limit is fixed by the lengths of entrance and run—the *wave-making features* of a ship.'

The laws of resistance given above relate only to smooth water, and do not take any account of the action of the wind and waves, which action, with the resultant pitching and tossing, will evidently cause the resistance of a ship in a seaway to be very different from that in smooth water; and from the nature of the case it is impossible to make a theoretical estimate of the extent to which the results may be thereby influenced. It is, however, clear from general observation and experience that length, size, and weight in ships tend to give them greater facilities of maintaining their speed in a seaway, and this is most conclusively shown by the regularity with which large ocean steamers make their voyages under all conditions of wind and sea.

Having thus briefly stated the leading points affecting the resistance offered to a ship moving through the water,

we now pass on to consider the methods by which it is overcome in steamships, and the action of the propellers employed. In order to make a complete investigation of the theory of propulsion, so far as it has yet been developed, it would be necessary to employ somewhat extensive algebraical reasoning which would be beyond the province of this treatise, so we shall confine ourselves to summarising, with only a very slight use of mathematical expressions, the leading principles and deductions that illustrate the action of a propeller in the water.

It is only within the last few years that the knowledge of the action of propellers has assumed a sufficiently definite form to deserve the name of a theory, and the progress that has been made in its development has been mainly due to the recognition of the importance of the principle of momentum; by which the effect of a force is estimated by multiplying it by the *time during which it acts* instead of the *space through which it acts*, as in the principle of work, which was explained in Chapter II. This principle and its application to the problems of propulsion were clearly explained by Professor Cotterill in papers contributed to Numbers 2 and 3 of the 'Annual of the Royal School of Naval Architecture and Marine Engineering.' The two principles are analogous and convertible, as may be seen from the following simple illustration; and we will point out the features in which they differ, and the reasons why the principle of momentum is the more suitable for solving the general problems connected with propulsion.

If a body move from rest in a straight line under the action of a constant force, P, then at the end of the time t

$$Pt = m\,v$$

where m = the mass of the body

and v = velocity in feet per second.

Mass.—The mass of a body is a constant quantity, depending on its size and density, and is equal to the weight of the body divided by the force of gravity. If D be equal to the weight of the body, and g represent the accelerating force of gravity,

$$\text{mass} = \frac{D}{g}.$$

For a cubic foot of sea water, which weighs 64 lbs.

$$\frac{D}{g} = 2 \text{ nearly.}$$

The product, $m\ v$, is called the momentum of the body; so that the force multiplied by the time through which it acts is equal to the momentum of the body; if the body had initially a given velocity, 'change of momentum' should be substituted for momentum.

If in the time t the body has moved through a space x, we have

$$Px = \tfrac{1}{2}\, m\ v^2,$$

where $\frac{1}{2}\ m\ v^2$ is called the $\frac{1}{2}$ vis-viva of the body, so that the force multiplied by the *distance* through which it acts (which is the *work performed*) is equal to the $\frac{1}{2}$ vis-viva generated; if the body had initially a given velocity the work done would be equal to one-half of the change in the vis-viva.

Therefore in the simple case of a constant force acting on a body, in a given direction, we see that if it be considered by the *time* during which it acts, its measure is the momentum or change of momentum produced; whilst if it be considered with respect to the *space* through which it acts, it should be estimated by the change in the $\frac{1}{2}$ vis-viva produced.

By comparing the two systems of measuring the effect of a force it may be seen how much more readily the principle of momentum may be adapted to the case of

propulsion than the principle of work, invaluable as the latter is in the case of forces acting on a rigid body.

Suppose a number of particles connected together in any manner and acted on by forces in any direction. The work done by these forces on the particles is expended in two ways.

1. Changing the vis-viva of the particles.
2. Overcoming their mutual actions.

The vis-viva is estimated by multiplying the mass of each particle by the square of its velocity, *without reference to its direction.* In a rigid body, or in a perfectly incompressible fluid, the second part of the action would not exist, and the work done by the forces would be entirely represented by the change in the $\frac{1}{2}$ vis-viva. If, however, we conceive the particles to be connected together by elastic bands, it is evident that the part of the work performed by the external forces, represented by the mutual actions between the particles, would be absorbed in stretching the strings. In an actual fluid, it is most improbable that the whole of the work of the forces would be utilised in the change of vis-viva, and a part of it must be assumed to be expended in mutual actions between the particles, and the difficulty of estimating this renders the principle of work not altogether suitable for application in many cases of hydro-dynamics. In the case of propulsion, the difficulties attending its application were so serious, that little or no progress could be made in the solution of the problem by its means.

By the application of the principle of momentum the question is much simplified. The momentum of a body, acted on by a force constant in magnitude and direction, is not like vis-viva independent of the direction of motion, but must always be estimated in the direction of the action of the force under consideration. The practical result

of this principle, in the case of a series of forces acting on a number of particles is, that the accumulated momentum *in any direction* is independent of the mutual actions between the particles; and if the forces have a constant resultant in any direction, the accumulated momentum, in any time, is equal to the product of that resultant into the time.

The advantages of the application of the principle of momentum to the case of propulsion are, therefore:

1. That it is true irrespective of mutual actions.

2. That it is not necessary to know the absolute motion of the particles, but only their resolved motion in any given direction.

On the other hand, it is important in applying the principle of momentum to consider *all the forces* acting on the system, and not merely those that do work; and this renders it less valuable in most mechanical problems than the principle of work.

In estimating the total momentum of a system the direction of motion must be considered; consequently, if two equal particles having the same velocity move in opposite directions, they neutralise each other, and their resultant momentum is zero. In cases of fluid motion, it often happens that the particles are moving in different directions with different velocities, so that whilst the total vis-viva may be great, the resultant momentum may be small. Hence, whatever may be the nature of the process by which work is dissipated in a fluid by *shock*, whether it be wholly due to infinitesimal whirlpools, or partly to mutual actions between the particles, there can be little doubt that in estimating the momentum of a fluid mass we need only concern ourselves with the visible motions generated in the water, without attempting to estimate the invisible motions, which cannot be measured but can only be determined empirically.

Hydraulic machines in general may be divided into two classes:

1. Those that operate by the weight and reaction of water.

2. Those that operate by the impact and reaction of moving water.

The theory of the action of propellers is very closely allied to that of the second class of hydraulic machines, and we shall probably be able to get a clearer conception of the subject by tracing the following steps in its consideration:

1. Impact of water on a fixed plane.
2. Reaction of water issuing from a fixed vessel.
3. Action of jet-propellers.
4. Action of feathering paddle-wheels.
5. Action of screw-propellers.

By the application of the principle of momentum, the pressure produced on a fixed plane by the impact of a jet of water striking it perpendicularly, as shown in Fig. 164, is easily ascertained. If we consider the motion of any particle, we see that at first it is moving with a velocity v perpendicular to the plane. As it approaches the plane its direction diverges, till after a certain time it is deflected by the plate and moves parallel to the surface of the plane, so that the momentum, perpendicular to the plane, becomes zero, being destroyed by the action of the plane. This is on the assumption that nothing in the nature of a rebound occurs, and that all the motion perpendicular to the plane is destroyed. This is true for all the particles in the jet, so that the effect of the plane is to

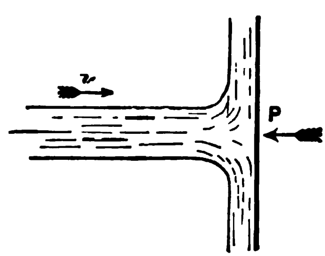


Fig. 164.

destroy the whole momentum of the jet, which originally had a uniform velocity v perpendicular to the plane.

If A = area of the jet in square feet, and v its velocity in feet per second,

$A\ v$ = number of cubic feet of water that strike the plane per second,

and if D represent the weight of a cubic foot of water,

$D\ A\ v$ = weight of water that strikes the plane per second.

The original momentum of the water delivered by the jet, in one second, perpendicular to the plane was

$$= \frac{D}{g} A\ v \times v$$

$$= \frac{D}{g} A\ v^2$$

This is entirely destroyed by the action of the plane, and consequently, if P be the total pressure produced on the plane,

$$P \text{ will be} = \frac{D}{g} A\ v^2.$$

It is evident that the greater the quantity of water acted on per second, the greater will be the pressure produced; and this principle is equally true with respect to the action of propellers.

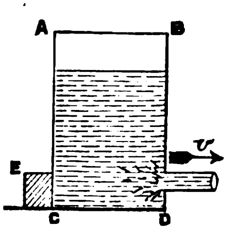


FIG. 165.

Next suppose a jet of water to be issuing from a vessel, A B D C (Fig. 165,) with a velocity v, through an orifice in the side; the vessel being kept from moving by the action of an obstacle C E; friction neglected.

We saw in the last example that the jet was capable of producing a pressure on a plane equal to the total momentum of the jet:

$$= \frac{D}{g} Av^2.$$

In the present instance, this momentum must evidently be balanced by the reaction of the obstacle to the movement of the vessel; or if R represent this reaction,

$$R = \frac{D}{g} Av^2.$$

When a vessel is propelled by the action of any propeller worked by forces within the ship herself, the total momentum of the sea and ship is always necessarily zero, and any forward momentum generated by the passage of the ship is exactly balanced by the backward momentum generated by the propeller; and it is only on this principle that the action of propellers must be explained.

Jet-propeller.—This is the simplest form of propeller, but one that has been rarely used, for reasons that we shall see farther on. It will, however, serve as a good illustration of the application of the principle of momentum to the investigation of the action of propellers in general. In this case, water is drawn from the sea by a pump through orifices in the bottom, and projected sternwards through orifices either at the sides or at the stern. The water acted on was originally at rest. It is drawn into the ship and consequently caused to move forward with a velocity V = the speed of the ship; and is lastly projected sternward with a velocity v, which we will suppose to be known.

The final velocity of the issuing water with respect to still water is therefore $= (v - V)$, and if A = the joint sectional area of the orifices, in square feet, the number of cubic feet of water acted on per second is $= Av$. The mass of water leaving the ship per second is therefore

$$= \frac{D}{g} Av,$$

and the total sternward momentum of the issuing jets is

$$=\frac{D}{g}Av\,(v-V).$$

From the principles previously stated, this must be equal to the sternward force produced, or, in other words, to the *thrust of the propeller.*

In the case supposed, in which the ship is moving in undisturbed water, this thrust is equal to the resistance of the ship, R, so that

$$R=\frac{D}{g}Av\,(v-V).$$
$$=2Av\,(v-V),\ \text{for salt water.}$$

It is evident that the thrust of a jet-propeller is theoretically independent of the position of the orifice, whether above or below the water level. If above, the pump would have to do some additional work in raising the water, and the opening would be dangerous in a war-ship; and if below, the resistance of the ship would be considerably increased by the dragging of the external pipes through the water.

The efficiency of the jet-propeller, if the loss from friction of the passages, shock, &c., be neglected, can be easily ascertained. The work done in forcing out the jets of water is equal to one-half the vis-viva generated; or,

$$=\frac{D}{2g}Av\,(v-V)^2.$$

This added to the useful work done in propelling the ship, R V, is equal to the total power expended; or,

$$\text{total work}=R\,V+\frac{D}{2g}Av\,(v-V)^2.$$

By substituting the value of R and reducing, this becomes

$$=\frac{D}{2g}Av\,(v^2-V^2).$$

The efficiency of the propeller

$$= \frac{\text{useful work}}{\text{total work}}$$

$$= \frac{\mathrm{R\,V}}{\frac{\mathrm{D}}{2g}\,\mathrm{A}v\,(v^2 - \mathrm{V}^2)}$$

$$= \frac{\frac{\mathrm{D}}{g}\mathrm{A}v\,(v - \mathrm{V})\,\mathrm{V}}{\frac{\mathrm{D}}{2g}\mathrm{A}v\,(v^2 - \mathrm{V}^2)}$$

$$= \frac{2\,\mathrm{V}}{v + \mathrm{V}}.$$

This is theoretically the maximum efficiency that any propeller can attain; as it is assumed that all the water is projected directly sternwards, and all the losses from friction of passages, shock, &c., are neglected. This becomes unity when $v = \mathrm{V}$, that is, when the speed of the jet is equal to that of the ship, so that the resultant velocity of the water, relatively to still water, is zero.

From the formula giving the thrust of the propeller, viz.,

$$\mathrm{R} = \frac{\mathrm{D}}{g}\mathrm{A}v\,(v - \mathrm{V}),$$

we see that the thrust of a propeller mainly depends on the product $\mathrm{A}\,v$. The smaller the value of v the greater must be the value of A; and since the efficiency becomes a maximum when $v = \mathrm{V}$, which is the smallest value v can have, it would point to the deduction that theoretically the larger A is made the more efficient would be the performance. In the limiting case when $v = \mathrm{V}$, $v - \mathrm{V}$ becomes = zero, and in this case A would become infinite. Generally, A is made as large as practical considerations will admit, so as to keep v, the speed of the race, as small as possible; the resultant sternward momentum of the

race with respect to still water representing a loss of thrust, and the higher its velocity the greater will be the loss from shock, &c. With reference to screw-propellers, as we shall see in a later portion of this chapter, this principle requires some modification under certain circumstances.

Feathering paddle-wheels.—We have taken the case of feathering paddle-wheels as our next step, as, in these propellers, the floats, when in the water, are supposed to be acting in a direct sternward direction, and to enter and leave the water normally, the area of the race of the propellers being equal to that of a pair of floats. In ordinary radial paddle-wheels there is a deal of local agitation and disturbance of the water, due to their oblique action on entering and leaving, which complicates the question, and the area of the race is not so clearly defined. The *race of a propeller* is the technical name given to the stream of water driven sternward by the propeller.

The water is assumed to be at rest before being acted on by the floats, and therefore, relatively to the ship, it would have a sternward velocity, V, equal to the speed of the ship.

If A= the area of a pair of floats, one on each side of the ship, the sternward momentum generated per second is

$$=\frac{D}{g}Av\ (v-V)$$

as before, and this is equal to the resistance of the ship R,

$$\text{or } R=\frac{D}{g}Av\ (v-V).$$

The work wasted in producing the race is equal to the ½ vis-viva generated;

$$=\frac{D}{2g}Av\ (v-V)^2;$$

$$=\frac{1}{2}R\,(v-V).$$

The total work done in propelling the ship is $=Rv$, and the useful work done is $=$ R V.

Therefore the total work wasted is equal to

$$R\,(v-V).$$

It will be seen from the above that the loss of work, due to the velocity produced in the propeller race, only amounts to one-half of the total power wasted, the remainder being absorbed in producing the violent churning and agitation of the water which is always produced by paddle-wheels. In practice the loss from this cause would be even more than one-half of the total power wasted, for in the above investigation we have neglected the resistance to forcing the floats in and out of the water, which considerably increases the work of the engines.

The efficiency of paddle-wheels, therefore, can never be greater than

$$\frac{RV}{Rv}, \text{ or } \frac{V}{v},$$

which may be taken to represent the maximum efficiency of these propellers.

Professor Cotterill, in a paper contributed to the 'Annual of the Royal School of Naval Architecture and Marine Engineering' for 1872, gives a rough comparison of the possible efficiencies of jet-propellers and feathering paddle-wheels. He assumes the enormous difficulties arising from large orifices in the ship's side overcome, so that the jet may send back the same quantity of water with the same velocity as the paddle-wheel, with 20 per cent. slip; for which purpose the orifices must be about the same size as the paddle-floats. The power wasted by the paddles will be 10 per cent. in the race, and from 25 to 30 per cent. in churning the water and in engine friction; whilst the jet-propeller will absorb 10 per cent. in the race and at least 35 per cent. in engine friction and

resistance in the passages. It will be seen even by this estimate that the performance of the jet is inferior to that of paddles, and it is clear that in general practice the area of the jet could not be made as great as that assumed, so that its speed must be increased in proportion as its area is decreased, which would make the comparison still more unfavourable to the jet-propeller. This defective action, due to the resistance of passages, &c., combined with the practical objections to the fitting of large orifices in the ship's side, either above or below the water, places the jet out of the region of practical propellers, except under very special circumstances.

The 'Waterwitch' is the only example of a ship with a jet-propeller in the Royal Navy, and the results of her trials sufficiently demonstrated the inefficiency of the system. With 760 indicated horse-power the 'Waterwitch' attained a speed of 9·3 knots, her displacement being 1,160 tons. The 'Viper,' twin-screw gun-vessel, of somewhat similar dimensions to the 'Waterwitch,' attained a speed of 9·6 knots with 696 indicated horse-power, her displacement at the time being 1,180 tons. The form of the 'Waterwitch' at the stern was much finer than that of the 'Viper,' which would considerably decrease her resistance. In comparing the performances of the two ships, a remarkable contrast between the quantities of water acted on by the two kinds of propellers will be seen. In the 'Waterwitch,' the joint area of the two nozzles was $5\frac{2}{3}$ square feet, and the quantity of water passing astern per second was about 150 cubic feet. The twin screws of the 'Viper' acted on over 2,000 cubic feet per second, or about fourteen times the quantity delivered by the jets of the 'Waterwitch.'

Further experiments with hydraulic propulsion were made in 1883. One of the second class torpedo-boats

built by Messrs. Thornycroft & Co. was fitted with a turbine instead of a screw propeller, and the greatest skill and care were exercised, both in the design of the boat and of her machinery, to insure the most efficient results. On the comparative trials between the hydraulic boat and the other boats of equal size fitted with screw-propellers, it was found that the speed of the hydraulic boat was no greater than could be attained in the screw boats with about half the power. The actual results were:

	I.H.P.	Speed	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
Hydraulic boat . . .	167	12·6	72
Screw-propeller boat . .	170	17·3	169

Full particulars of the design and performance of this boat are contained in a paper by Mr. Sydney W. Barnaby, read at the Institution of Civil Engineers on February 26th, 1884.

Radial paddle-wheels.—In paddle-wheels with radial floats, it is evident that only the float at the bottom of the wheel at any time is vertical, and giving direct sternward velocity to the water. All the others act obliquely on the water and have a vertical as well as a horizontal reaction, the former, although absorbing a large proportion of the power of the engines, being entirely wasted, so far as propulsive effect is concerned. The greater the immersion of the wheel, the greater will be the loss due to the vertical component of the pressure on the float. These wheels, therefore, should be so designed that at the maximum load draught of the ship, they should not be immersed more than one-quarter the diameter of the wheel; for beyond this limit the loss from the vertical reaction increases at a very rapid rate.

With these propellers it is almost impossible to determine, with any degree of accuracy, the area and speed of the race. It is frequently assumed, for practical purposes,

that the area of the race is equal to the length of the float multiplied by the maximum depth of immersion of its lower edge, and that its speed is that of the outer periphery of the wheel. This is probably useful for comparison of the efficiencies of different wheels of the same class, but it is too obviously incorrect to have any value in a scientific investigation of the propulsive effect of the propellers. The average speed of the race would be less than that assumed, and would approximate more nearly to that of the centre of pressure of the floats. This, however, is difficult to calculate with exactness, and unnecessary, as in any case the results could only be used for comparison and would not give the actual thrust of the propeller, as the effective area of the race cannot be defined.

The chief objection to the employment of paddle-wheels for ocean navigation arises from the practical difficulties attending the variation in immersion during long voyages, owing to the lightening of the ship by the consumption of coals, stores, &c. Even with feathering wheels, in which the floats are approximately vertical when in the water, the loss from forcing the floats in and out of the water, churning, &c. is very much increased when the wheels are deeply immersed; and it is evident that if the wheels are to be sufficiently immersed at the end of a long voyage they must have been too deep in the water on starting. For short voyages, in which the draught of water is comparatively unchanged, paddle-wheels may be advantageously employed, as has been shown by the Holyhead mail packets and the London and North-Western passenger steamers, which steam at rates of 18 to 20 knots per hour, and make their passages between Holyhead and Dublin with remarkable regularity. Paddle-wheels are almost essential for propulsion in many shallow rivers, where the depth of water is insufficient to admit the use of screws. For ocean navigation, however, in addition

to the loss of efficiency from alteration in immersion, paddle-wheels are objectionable in consequence of the racing and straining of the machinery due to rolling motion in a seaway, and consequently for this purpose they have been entirely superseded by the screw-propeller. It also appears probable that, for most other purposes, they will be displaced by the screw, and before long the paddle-wheel as a propeller will only be employed in cases in which special circumstances forbid the application of screw-propellers.

Screw-propeller.—The action of a screw-propeller is much more complex than that of the two kinds of propellers previously discussed. This complication is due mainly to the following causes:—

1. The action of the propeller on the water is oblique instead of direct.
2. The velocities of the several particles of water acted on by the screw are different from each other.
3. The screw always acts on water that has previously been set in motion by the ship.

The difficulties attending an exact investigation of the action of screw-propellers are consequently so great, that hitherto the problem cannot be said to have been completely solved; but we will endeavour to explain as clearly as may be possible, without the use of mathematics, the principles involved in the consideration of the action of a screw-propeller, so that a true conception may be formed of the nature of its performance and the points on which its efficiency depends. Oblique action, other things being equal, is always a cause of loss of efficiency in a propeller, and the fact that, in spite of this, the screw is a practically efficient propeller, is explained by the circumstance that it operates upon a much greater quantity of water than could be acted on by a pair of paddle-floats, or by any other propeller in a ship of the same size. Area for area, the

screw would probably be found to be less efficient than the paddle-wheel; but the efficiency of a propeller depends to a great extent on the quantity of water acted on, and the screw is the form of propeller that is best adapted to fulfil this condition.

Before entering further into the discussion of the action of screw-propellers, it will be desirable to define a few of the technical terms that will frequently be used.

Diameter.—The diameter of the screw is the diameter of the circle formed by the tips of the blades when revolving. The area of this circle is called the *disc area* of the screw.

Pitch.—The pitch of the screw is the distance through which the screw would advance *in one revolution*, provided it turned in a solid nut.

Speed of screw.—The speed of the screw is the distance it would advance *in a unit of time*, supposing the screw to be working in a solid nut. This is obviously equal to the pitch of the screw multiplied by the number of revolutions made per unit of time, say per second, or per minute as the case may be.

Slip.—In consequence of the screw-propeller working in a yielding medium, the ship generally does not advance at as great a rate as the speed of the screw; and the difference between the speed of the screw and the speed of the ship is called the slip of the screw.

If v = speed of the screw-propeller.

V = " " ship.

$v - V$ = actual slip of the screw.

$$\frac{v - V}{v} = \text{slip of the screw expressed as a fraction.}$$

$$\frac{v - V}{v} \times 100 = \text{percentage of slip.}$$

This, however, is only the *apparent slip of the screw*, and the difference between this and the *real slip of the*

screw must be clearly borne in mind. The *apparent slip* and the *real slip* can only be identical in the event of a screw acting on water previously at rest, which can never be the case with the water operated on by the screw-propeller of a ship. The passage of the ship through the water causes a wake to follow her, so that the screw-propeller always acts on water that has previously been set in motion by the ship. The velocity of this stream must therefore be considered with the *apparent slip* in order to obtain the *real slip*, which represents the true value of the backward velocity impressed on the water by the propeller. The speed at which the water follows the ship depends on her form, and is difficult to ascertain readily, so that the slip usually calculated and inserted in tables is the apparent slip only and not the real slip.

Negative slip.—From the nature of the medium in which the screw-propeller works, it is clear that in every case there must necessarily be *positive real slip*; but it is not difficult to imagine cases in which the body of water following the ship had such an initial velocity that the *apparent slip* of the screw might be *negative*. This phenomenon of negative slip—that is, of the speed of the ship being actually greater than that of the screw-propeller—has sometimes occurred, and before the true theories of resistance and propulsion of ships were sufficiently developed, was almost beyond explanation, giving rise to a vast number of ingenious theories to account for its existence. It is, however, quite clear, from our present knowledge of the subject, that the fact of the screw working in the following wake will account for the phenomenon to a large extent at least.

Whenever screws act on water previously set in motion by the ship, there is always a loss of thrust, because the increased thrust, due to the forward velocity of the water in which the screw acts, is always more than counter-

balanced by the diminution in the pressure of the water under the stern of the ship, produced by the action of the propeller. Negative slip therefore always means a loss of efficiency, and the apparent slip of a screw calculated from the trials of a ship is, of itself, no fair criterion of the true efficiency of the performance of the ship.

We would observe in passing that possibly in some cases, in which the slip was apparently negative, the result may be attributed, to some extent, to the difficulty of correctly estimating the true mean effective pitch of an ordinary screw-propeller. The method adopted practically is to divide each blade into a number of equal sections, and to measure the pitch or pitches of each division separately. It is generally found that the pitch is somewhat different in the several divisions; and frequently in each section the pitch of the leading part differs from that of the following part of the blade. The arithmetical mean of all the pitches thus measured, for all the screw-blades, is found, and called the mean pitch of the screw; but considering the different velocities with which the several sections pass through the water, and their different obliquities and areas, it is by no means certain that this method gives the true mean pitch of the propeller as regards its propulsive effect; and a comparatively small error in estimating the mean pitch, might considerably affect the calculated apparent slip of the screw.

The laws of slip as deduced from experiment may be thus summarised:—

1. For the same screw the slip increases with the resistance of the vessel.

2. The apparent slip increases slightly as the number of revolutions of the screw per minute is increased.

3. Other things being the same, the slip increases with the pitch. Hence, neglecting friction, screws of fine pitch revolving at great speed would be most efficient.

4. The slip increases in proportion as the relative resistance, or $\frac{\text{midship section}}{\text{disc area}}$, increases. It is therefore generally desirable to make the diameter of the screw-propeller as large as possible.

5. The slip decreases as the area of the screw-blades, or, what amounts to the same thing, the length of the screw, increases. This is only true within certain limits. The slip decreases rapidly with increase of length up to the point at which the aggregate length of the screw-blades is 30 per cent. of the pitch. For increases of length between 30 and 75 per cent., the slip diminishes slowly, and for increases of length beyond 75 per cent. the diminution of slip becomes insensible. The number of blades has very little influence on the slip when their aggregate length is the same.

The action of the screw-propeller is to drive sternward a cylindrical column of water, usually called the propeller race, and the thrust of the screw is measured by the sternward momentum generated in this race in a unit of time. The area of this race is approximately equal to that of the screw disc, less that of the boss of the screw: the race being in fact a hollow cylindrical column. In consequence of the obliquity of the propelling surfaces, the race receives a rotatory as well as a sternward motion; and this centrifugal action causes a certain loss of thrust. The true conception of the race of a screw-propeller is that of a series of concentric cylinders of water moving sternward, and rotating, at different velocities; and it is evident that the thrust must be to some extent diminished both by the centrifugal motion and by the frictional action of the particles of water.

The most important feature in the action of a screw-propeller, as affecting its efficiency, is the effect it produces on the water under the stern of the ship. In the absence

of the propeller the water displaced at the bow by the passage of the ship would tend to close in under the stern, and cause a forward pressure there. The action of the screw withdraws this water, and consequently diminishes the pressure of water under the stern, which is equivalent to increasing the resistance of the ship. The *resistance augmentation*[1] in ordinary single screw ships, due to this cause alone, was estimated by the late Mr. Froude to be from 40 to 50 per cent. of the natural resistance of the ship, or the resistance experienced by the ship when towed at the same speed. The experiments were, however, made on a ship with a full run and very thick stern and rudder posts, and a large percentage of the increased resistance of the ship experimented on might be accounted for by the action of these posts alone. It would appear that in a well-formed single screw ship, except for the effect of the stern and rudder posts, the *resistance augmentation* due to the action of the screw would be generally from 15 to 18 per cent. of the ship's natural resistance. In the 'Iris,' and other twin screw ships of similar form, the increase is only 10 to 12 per cent., and in some cases it has been found to be as low as 8 per cent.

To a greater or less extent, however, augmented resistance due to the action of the screw-propeller must always occur when the screw is placed close to the stern of the ship. It may be very considerably reduced by placing the screw some distance behind the ship, and this explains to some extent the very high speeds attained by some fast torpedo-boats and other similar vessels. Mr. Froude

[1] Mr. R. E. Froude, in his paper on Screw Propeller Efficiency, read at the Institution of Naval Architects in 1883, pointed out that this *resistance augmentation* was essentially a function of the thrust of the propeller, and should be regarded as a percentage drawback or discount from the generated thrust whereby the thrust available for overcoming the natural hull-resistance is less than the thrust actually generated. He therefore considered it more accurate to term it *thrust deduction* instead of *resistance augmentation*.

proved by experiment that if a screw were placed, from one-third to one-quarter of the extreme breadth of the ship, clear from the stern, the increase of resistance due to its action was only one-fifth of that ordinarily produced. It is, however, clear that a plan of this sort could not be adopted in a sea-going vessel, for in addition to the mechanical difficulties of supporting the propeller in that position, there are the insuperable objections of want of protection and greater liability to fouling and damage. For the present, therefore, at all events, we must submit to a loss of efficiency in the screw-propeller, due to the *increase in the resistance* of the ship, and must endeavour to reduce it to a minimum by making the run of the ship as fine as possible, and by adopting all other suitable means to cause a free flow of water to the screw.

With screw-propellers revolving rapidly, it is evident that there must be a considerable waste of power in overcoming the edgewise and frictional resistance offered by the water to the motion of the screw. The power expended in this work cannot be estimated at less than 4 per cent. of the total indicated horse-power developed by the machinery.

Notwithstanding these defects the screw-propeller has proved itself practically to be the most efficient propeller hitherto used, which, as before stated, is explained by the circumstance that it is able to operate on much larger quantities of water than any other propeller. It has entirely superseded the paddle-wheel for ocean navigation, as it is very slightly affected by the two causes which have such a serious prejudicial effect on the efficiency of paddle-wheels, viz. variation of immersion, and rolling in a sea-way. Sail-power can also be much more readily utilised, if required, in a screw than in a paddle-wheel steamer; and, in a war-ship, the paddle-wheels being so much exposed would be very liable to danger in action, whilst the

screw-propeller would be protected by being under the stern of the ship and below the surface of the water.

The general considerations relative to the action of screw-propellers point to the conclusion that the greater the area and the smaller the pitch the greater will be the efficiency. This is generally true, but it is subject to modification in practice. It is most important that the highest part of the screw-blades should always be a sufficient depth below the surface of the water to prevent the mixing of air to any considerable extent with the propeller race, which would decrease the quantity of water acted on by the screw, and diminish its thrust. As the lowest point of the screw-blade must, in ordinary ships, be above the keel, this consideration limits the maximum diameter that can be advantageously given to the screw. Again, if a fine pitch be adopted, the slip is reduced, and consequently the loss from that cause; but, on the other hand, since the revolutions of the engines must be increased in order to maintain the same speed of ship as with a screw of coarser pitch, the work expended in overcoming the frictional and edgewise resistance of the screw is much greater, and the loss from this cause consequently increased. Mr. Froude in his paper on Pitch, Slip, and Propulsive Efficiency, published in the 'Transactions of the Institution of Naval Architects,' for 1878, estimates the work absorbed in overcoming the frictional resistance alone of the outmost foot of a screw-propeller, 20 feet in diameter, making 80 revolutions per minute, at 120 indicated horse-power. These facts, therefore, considerably modify the theoretical conditions relative to the most efficient diameter and pitch of screw-propellers.

Practically, with single screws, it is desirable that the diameter should, as a general rule, be made as large as is consistent with keeping the upper edge sufficiently immersed; and the pitch is usually from one to one and a

half times the diameter, which ratio has been found, by experiment, to give the most efficient results. The disc area in ordinary screw ships varies from one-half to one-quarter of the immersed midship section of the ship, one-third being a good average value.

The most important consideration relative to the efficiency of a screw-propeller is the facility offered for the free and unrestricted flow of a plentiful supply of water to be operated on; and this is attained by making the after run of the ship with as fine lines as possible. This fact was not fully understood when screw-propellers were first introduced, but fineness of run is now recognised as a necessary feature in the construction of screw ships. Mr. Arthur Rigg and Mr. Griffiths, who have devoted great attention to screw-propulsion, proposed the fitting of guide plates or vanes to induce a current of water to flow to the screw-propeller. This system has been carried out with great success by Mr. John I. Thornycroft in some steamers of shallow draught, which were described in a paper read at the Institution of Naval Architects, in March 1885.

Twin screws.—In cases where the draught of water of the ship has not been sufficient to allow a single screw to be used of diameter great enough to transmit the full power of the machinery, two screws have been employed, one on each side of the ship, with very satisfactory results. The area of the propeller race has by this means been much increased, and it would appear that from their positions under the quarters of the ship, the water flows to them more readily, and they do not increase the resistance of the ship to as great an extent as the single screw, by the diminution of the pressure of the water under the stern. The trials of single and twin screw ships in the Royal Navy, where alone the latter system has been tried to any considerable extent, show that the propulsive efficiency of twin screws is greater than that of single screws.

The duplication of the machinery in twin screw ships also prevents a total collapse in case of accident; for if one set of engines broke down, the other set would be capable of propelling the ship at a fair rate of speed, so that she might safely and speedily reach a port to have her defects made good. By the use of twin screws the manœuvring power of the ship is greatly increased, for by working one screw ahead and the other astern, the ship may be turned in her own length without necessitating any speed of ship, and this would be of great importance in case of action. The ship could also be steered by the propellers alone, independently of the use of the rudder if necessary in case of accident. The mechanical arrangements in a twin screw ship enable the hull to be subdivided into longitudinal water-tight compartments, by fore and aft bulkheads, separating the respective engine and boiler rooms, which adds greatly both to the strength and safety of the ship. There are many other advantages resulting from the adoption of twin screws, which are, however, sufficiently obvious on consideration to prevent the necessity of further enumeration.

Multiple screws.—In some special ships, such for example as the Russian circular ships or *cyclads*, multiple screws have been used. In these ships, six screw-propellers are fitted, three on either side of the centre line, each driven by separate engines. It was necessary to resort to some such method as this in order to obtain the necessary propeller disc area with the limited draught, but it is probable that their performance would not be very efficient, in consequence of the action of the several screws interfering to a greater or less extent with the flow ot water to the adjoining screws, probably preventing them from getting a sufficient supply of water and thereby diminishing their thrust. This plan has only been tried to a very limited extent, and while it is desirable to

call attention to it, it is unnecessary to devote any further space to its discussion at present.

Some very important facts relative to the action and propulsive efficiency of twin screws were obtained from the steam trials of Her Majesty's despatch vessel 'Iris,' fitted with twin screws, and originally intended to attain a speed of from 17 to 17½ knots an hour, with a displacement of about 3,700 tons. It was estimated that for this speed the engines would have to develope about 7,000 indicated horse-power. The screws fitted in the first instance were each four-bladed, 18 ft. 6½ in. in diameter, having a mean pitch of 18 ft. 2 in.; the diameter being made as large as possible in accordance with the generally accepted theory with reference to disc area, which, though substantially correct, would appear to require some modification for twin screws. The projected area of the blades, that is their area projected on an athwartship plane, was made about three-tenths of the whole disc area which is the usual proportion for four-bladed screw-propellers.

On the first official trial, which was made on December 14th, 1877, the engines made an average of 90·318 revolutions per minute, developing a mean indicated horse-power of 7,060. The speed of the ship, however, was only 16·4 knots per hour, or about one knot less than was anticipated. This great and unusual difference between the estimated and actual performances of the ship, caused the whole case to be carefully investigated, and trials were made under various conditions, in order to discover and remedy the defects if possible, and from these trials most valuable results were obtained, and a much clearer light thrown on the problems connected with twin screw propulsion. A full discussion of the case is given in an important and valuable paper contributed by Mr. J. Wright, C.B., Engineer-in-Chief of the Navy, to the Institution of Naval Architects in 1879, which is published

H.M.S. 'Iris'—Experimental Trials.

	First series						Second series				Third series					Fourth series				
	1	2	3	4	5	6	1	2	3	4	1	2	3	4	5	1	2	3	4	5
Date of trial . . .	Feb. 4, 1878				Feb. 5, 1878		Feb. 15, 1878				July 3, 1878					Aug. 1, 1878				
Draught of water { forward	15 ft. 7½ in.						15 ft. 7½ in.				15 ft. 8 in.					15 ft. 8 in.				
Draught of water { aft	20 ,, 6½ ,,						20 ,, 6½ ,,				20 ,, 6 ,,					20 ,, 6 ,,				
Immersed midship section	700 sq. ft.						700 sq. ft.				700 sq. ft.					700 sq. ft.				
Displacement . . .	3,290 tons						3,290 tons				3,290 tons					3,290 tons				
Revolutions per minute .	91·04	82·148	65·105	43·358	24·25	18·0	88·89	81·175	65·07	45·075	97·189	85·388	61·343	40·963	21·6	93·25	76·93	59·385	39·15	19·26
Indicated horse-power .	7,503	5,251	2,560	755	217	143	4,368	3,306	1,637	571	7,714	5,108	1,833	606	184	7,556	3,958	1,765	596	163
Speed in knots . . .	16·577	15·123	12·064	8·187	—	—	15·726	14·517	11·58	7·5	18·573	16·564	12·279	7·797	—	18·587	15·746	12·475	8·321	—
$\frac{(\text{Speed})^3 \times \text{mid. sect.}}{\text{I. H. P.}}$ . .	425·0	461·1	480·1	506·6	—	—	623·3	647·8	664·0	616·0	581·4	628·8	707·0	547·5	—	594·9	690·5	770·0	676·7	—
$\frac{(\text{Speed})^3 \times (\text{displ}^t.)^{\frac{2}{3}}}{\text{I. H. P.}}$ . .	134·3	145·7	151·7	160·1	—	—	197·0	204·7	209·8	194·6	183·7	196·8	223·4	173·0	—	188·0	218·2	243·3	213·8	—
Screw-propellers: Description . . .	Common with four parallel blades						Same screws as before, with two alternate blades removed from each				Modified Griffiths—four blades					Modified Griffiths—two blades				
Screw-propellers: Diameter . . .	18 ft. 6½ in.						18 ft. 6½ in.				16 ft. 3½ in.					18 ft. 1½ in.				
Screw-propellers: Mean pitch . . .	18 ,, 2 ,,						18 ,, 2 ,,				19 ,, 11½ ,,					21 ,, 3½ ,,				
Screw-propellers: Slip per cent. . .	1·57 neg.	2·69 neg.	3·36 neg.	5·33 neg.	—	—	1·99	0·93	1·41	2·29	2·97	1·50	1·63 neg.	3·36	—	5·04	2·49	0·07	1·25 neg.	—
Screw-propellers: Expanded area of all the blades in square feet	194·4						97·2				144·0					112·0				
Screw-propellers: Projected area of all the blades in square feet	160·0						80·0				120·0					98·0				
Screw-propellers: Ratio of projected area of blades to disc area of propeller	0·296						0·148				0·288					0·190				

in vol. xx. of the Transactions of the Institution. A table which was attached to Mr. Wright's paper, showing the results of the several trials, is reproduced here for information and reference.

The main facts are as follow :—

Four series of trials were made with different screws, and at different rates of speed. From the results thus obtained the curves of indicated horse-power and indicated thrust were drawn, showing the laws of variation of power and speed with the different screws. The method of constructing these curves is explained in Chapter XXI. For our present purpose it will be sufficient to confine attention to the maximum power and speed in each of the four series of trials.

In the first series the original four-bladed screws of 18 ft. 6½ in. diameter were used. The maximum speed attained was 16·5 knots, the engines making at the time 91 revolutions per minute and developing 7,503 indicated horse-power.

The second series was made with the same screws as before, but with two alternate blades taken off each, thus converting them into two-bladed screws of the same diameter and pitch as the original four-bladed screws. The results from this series of trials were very remarkable, and pointed to the fact that the original area of the propellers was too great. As these screws, with two blades removed, were not considered strong enough to transmit the full power of the engines, the highest power at which the engines were worked was 4,368 indicated horse-power, which was obtained with 88·89 revolutions of the engines. The speed of the ship was 15·726 knots, or only eight-tenths of a knot less than was previously obtained with 7,503 indicated horse-power. With the original four-bladed screw-propellers a speed of 15·123 knots required the exertion of 5,250 indicated horse-power, and 4,368 indi-

cated horse-power would only have driven the ship at the rate of about 14½ knots per hour.

This trial therefore showed most conclusively the important fact, that the speed of the ship might be very considerably increased by a modification of the form and dimensions of the screw-propellers. It was not, however, sufficient to show that the two-bladed screw was inherently superior to one with four blades, and the vibration from the two-bladed screw was objectionable.

The third series of trials was therefore made with modified four-bladed screw-propellers, the diameter being reduced to 16 ft. 3½ in., which was 2 ft. 3 in. less than the original propellers, and the outer edges of the blades were considerably rounded; the mean pitch was increased by 1 ft. 9½ in. making it equal to 19 ft. 11½ in., and the projected area of the blades was 0·288 of the total disc area. These screws, working a little over 97 revolutions per minute, propelled the ship at a speed of 18·573 knots per hour, the engines developing 7,714 indicated horse-power. The vibration in the ship was very moderate. This is a remarkable performance, being two knots more than was obtained with the original screws, with an increase of only 211 indicated horse-power. It must be borne in mind that the only alterations made were in the form and dimensions of the propellers, everything else being practically the same as on the original trials.

The fourth series of trials was made with two-bladed propellers, constructed of sufficient strength to transmit the full power of the engines, and the results, as regards speed, were slightly better than with the modified four-bladed propeller. The diameter of the screws in this case was 18 ft. 1½ in., mean pitch 21 ft. 3½ in., and the projected area of the blades, 0·19 of the complete disc area. The highest speed obtained was 18·587 knots, with 7,556 indicated horse-power, the engines making 93·25

revolutions per minute. The vibration was greater than with either of the four-bladed screws.

The comparative inefficiency of the original screws was at first supposed to have been chiefly due to the large amount of power absorbed in overcoming the frictional and edgewise resistance of the propellers. Mr. Wright, however, showed very clearly that the whole of the loss could not be attributed to this cause, and that a very large proportion of the total loss of efficiency was due to the following causes:—

1. The four broad-pointed blades in the original propellers were probably very fruitful in eddy-making; and as the blades followed each other, each blade would act on eddies set up by the preceding blade, particularly at the points where they were broadest and their velocity greatest. The efficiency would therefore be diminished, both by the eddy-making itself, and by the diminished thrust resulting from the disturbance of the water on which the propeller-blades operated.

2. The four large broad-pointed blades of the original propellers, following each other so rapidly, and being in such close proximity to the sides of the ship, would necessarily interfere much more than the other screws with the stream line motion of the water closing in under the stern, and would thus cause a greater *augment of resistance* than in the other cases.

CHAPTER XXI.

COEFFICIENTS AND CURVES OF PERFORMANCE.

Coefficients of performance.—For many years at the Admiralty the following coefficients have been used to indicate the resistance and propulsive efficiency of ships belonging to the Royal Navy.

Let A = area of immersed midship section of the ship in square feet.

D = displacement of the ship in tons.

V = speed of ship in knots per hour.

Then,

$$1\text{st coefficient} = \frac{A\,V^3}{\text{I.H.P.}}$$

$$2\text{nd coefficient} = \frac{D^{\frac{2}{3}}\,V^3}{\text{I.H.P.}}$$

These are based on the assumption that the resistance offered by the water to the motion of the ship varies as the square of the speed, and that, consequently, the power required to overcome this resistance would vary as the cube of the speed. This is only true for moderate speeds, but the coefficients calculated from these formulæ have been very useful for comparing the relative performances of ships, especially when somewhat similar in form, and have been valuable, in the case of a new design, as data for approximating to the indicated horse-power necessary to drive the ship at an assumed maximum rate of speed. On the whole, the second coefficient, in which the efficiency is referred to the two-thirds power of the displacement,

has been found to be the more trustworthy, giving a fairer measure of the resistance than the midship section coefficient, especially in dealing with ships that are not similar in form.

These coefficients, however, though they have done good service, and were for many years the only records of the efficiency of performance of steam-ships, are not sufficiently accurate for scientific purposes, and fail to indicate many points of importance, which are shown by more correct and scientific methods of recording and analysing the results of steam trials of ships.

Curves of indicated horse-power.—It has for many years been a well-known fact that, at the higher rates of speed, the resistance of ships often varied at a very much higher power of the speed than the square, and Messrs. Denny, of Dumbarton, originated a more accurate and instructive method of conducting and recording the results of steam trials. Instead of trying the ships on the measured mile at full and half powers simply, they carried out, in several ships, trials at several different rates of speed, varying from the maximum to speeds as low as from three to four knots per hour, so that the indicated horse-powers required for the respective speeds could be ascertained and tabulated. In order to render the results more palpable and suggestive they were shown in a graphical form, by the construction of diagrams, in which the speeds were set out as abscissæ and the corresponding horse-powers as ordinates. Having thus determined a sufficient number of points, a fair curve was drawn through them, and the indicated horse-power, corresponding to any intermediate speed, could then be ascertained by simply drawing the vertical ordinate from the point representing the speed required.

The details of construction of these curves of indicated horse-power is shown in Fig. 166. In this case the

speeds at which the ship was tried were those marked A, B, C, and D, and the corresponding horse-powers were shown on the scale of indicated horse-power by the points E, F, G, and H. By drawing the dotted horizontal and vertical lines as shown in the diagram, the points P, Q, R, and S are obtained. It is clear that when the horse-power is zero the speed will also disappear; so that if a fair curve be drawn through the points P, Q, R, and S, to touch the horizontal base line at the origin, where the

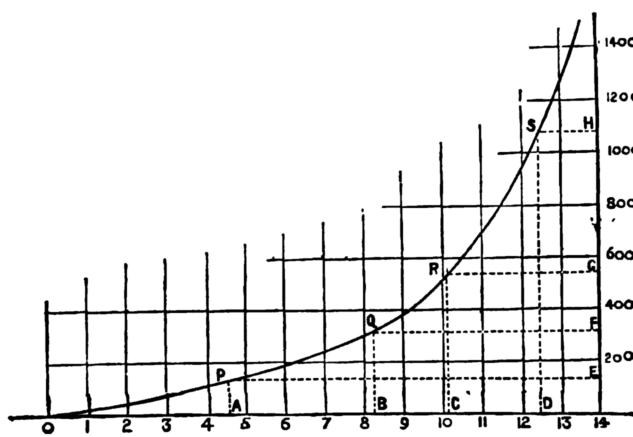


FIG. 166.

speed is = zero, it will represent the varying relations between the indicated horse-power and the speed of the ship; or in ordinary language, the curve of the indicated horse-power of the ship.

When this curve is drawn, the horse-power required to drive the ship at any assumed speed may be ascertained, by drawing the ordinate from the base line to the curve, at the point representing the assumed speed on the given scale. From these diagrams many important facts were learnt, and they must be regarded as the inauguration of

a system of recording the results of steamship performance, from which the possibilities of acquiring knowledge of defects, and of initiating improvements, are very great.

Curves of indicated thrust.—In 1876 the late Mr. Froude read a paper at the Institution of Naval Architects, in which he proposed an extension and improvement of Mr. Denny's system, by the substitution of the indicated thrust of the propeller for the indicated horse-power, as the ordinates of the diagram. From the curves of indicated thrust most valuable results have already followed, the most important probably being that the indicated thrust due to the initial or constant friction of the machinery may be determined by a simple geometrical construction. As these curves of indicated thrust are now used in her Majesty's service, and will without doubt be more fully developed and generally applied in future, it will be well to devote a few lines to the explanation of their construction and application.

The indicated horse-power of itself is not altogether reliable as a measure of the propulsive efficiency, as it combines in one item the performances of the ship, machinery, and propeller. Mr. Froude, therefore, deemed it desirable to reduce the indicated horse-power to a force factor, by dividing it by the speed of the propeller, the result being what he termed *indicated thrust.*

The indicated thrust may be estimated in either of two ways:—

1st. By multiplying the mean effective pressure on the piston in pounds per square inch by twice the length of the stroke in feet, and dividing the product by the pitch of the propeller in feet.

2nd. By multiplying the actual indicated horse-power by 33,000 to bring it to foot-pounds, and dividing the product by the pitch of the propeller in feet multiplied by

the number of revolutions per minute; which is the speed of the propeller in feet per minute.

If p_m = mean effective pressure on the piston in pounds per square inch

S = length of stroke in feet

P = pitch of propeller in feet

N = No. of revolutions per minute

Indicated thrust is

$$= \frac{p_m \times 2S}{P},$$

$$\text{or} = \frac{\text{I.H.P.} \times 33{,}000}{P \times N};$$

and either of the above formulæ, as may be most convenient in any given case, may be used in the calculations.

Having calculated the indicated thrusts for the trial speeds of the ship, a curve is constructed, with the speeds as abscissæ, as in the indicated horse-power curves, but with the indicated thrusts as ordinates instead of the indicated horse-powers.

Fig. 167 is an example showing the construction of the indicated thrust curve. Vertical ordinates, equal to the indicated thrusts at the respective speeds, are set up, and a series of points, *a*, *b*, *c*, *d*, and *e*, are thus obtained, through which a fair curve is drawn.

The indicated thrust is composed of several elements, which may be thus enumerated.

1. Useful thrust, or ship's true resistance.

2. Augment of resistance, due to the action of the propeller in diminishing pressure under stern of ship.

3. Equivalent of friction and resistance of the screw-blades in their edgewise motion through the water.

4. Equivalent of friction due to the dead weight of the working parts, piston packings, &c., which constitutes the initial or slow speed friction of the engines.

5. Equivalent of friction of the engines due to the working load.

6. Equivalent of the air-pump and feed-pump duty, &c.

It is probable that items 2, 3, and 5 are all very nearly proportional to the useful thrust. Item 6 is probably

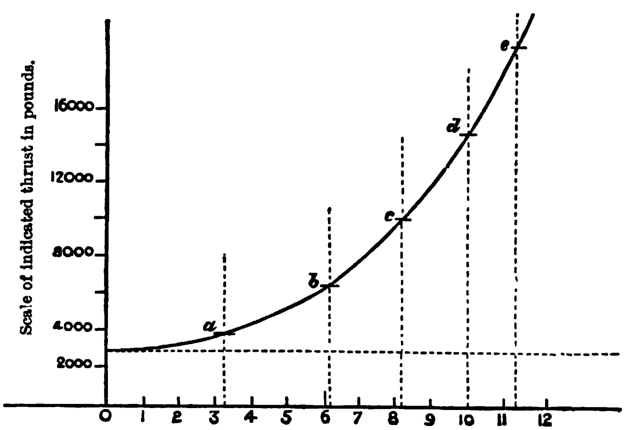


FIG. 167.

nearly proportional to the square of the number of revolutions, and thus, at the lower speeds at least, to the useful thrust. Item 4 probably remains constant at all speeds, and for convenience may be regarded as constant.

Mr. Froude showed graphically how the equivalent of the initial or constant friction may be separated from the indicated thrust throughout, so that the remainder, which shows the thrust approximately proportional to the ship's true resistance, may be ascertained. It will be observed, on drawing the thrust curves, that they do not descend to the thrust zero when the speed disappears, but tend to cross the vertical axis at some distance above it, representing a considerable amount of thrust at the zero of

speed. This apparent thrust when the speed is reduced to zero, and when consequently there can be no real thrust, will represent the thrust equivalent to the initial or constant friction of the engines; and if a horizontal line be drawn through this point of intersection, the height thus cut off from the thrust ordinates would represent the deduction to be made from them in respect of the constant friction, and the remainder of the ordinates between the

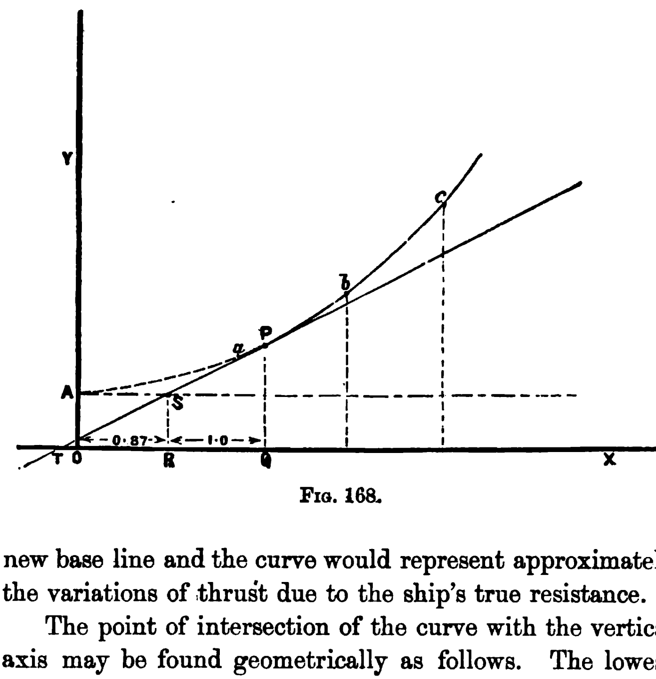


Fig. 168.

new base line and the curve would represent approximately the variations of thrust due to the ship's true resistance.

The point of intersection of the curve with the vertical axis may be found geometrically as follows. The lowest point in the curve found by trial should, if possible, correspond to a speed of three or four knots. For these speeds the resistance has been found to vary as the 1·87 power of the speed.

In Fig. 168 let *a b c* represent the lowest portion of the thrust curve corresponding to speeds of about from

3 to 6 knots, say. From any point P near the lower end of the curve, draw the tangent P T. Divide the abscissæ O Q at the point R in the ratio 0·87 to 1, so that O R is to R Q as 0·87 is to 1, and through R draw the ordinate R S cutting the tangent in S. If a horizontal line be drawn through S, cutting the vertical axis O Y in A, A will be the point through which the thrust curve will pass, and O A will represent the thrust equivalent to the initial or constant friction of the engines. For practical purposes no material error would be introduced by bisecting O Q instead of dividing it in the ratio of 0·87 to 1, which would simplify the process somewhat, and be very convenient in many cases on shipboard, as will be presently pointed out.

From experiments made by Mr. Froude he concluded, that, generally, in ships of ordinary form, only from 37 to 40 per cent. of the total power exerted by the engines is utilised in the useful thrust of the propeller, and he quantified the various items of the total distribution of the power in the following manner:

The several components of the horse-power are taken in the same order as the items of indicated thrust given on p. 420.

1. Horse-power due to ship's net resistance, or effective horse-power; represented by E.H.P.

2. Horse-power due to augment of resistance. This was estimated as = 0·4 E.H.P. As pointed out at p. 406, this estimate is probably high for well-formed ships.

3. Horse-power due to friction of screw-blades = 0·1 E.H.P.

4. Horse-power due to initial or constant friction is taken at 0·143 of ship's horse-power, = 0·143 S.H.P. By the ship's horse-power is meant the horse-power equivalent to the actual speed of the ship, neglecting that lost by the slip of the propeller.

5. The horse-power due to the friction from the working load is, at full power, at least equal to the horse-power due to initial friction, and is taken $=0{\cdot}143$ S.H.P.

6. The horse-power required for working the air and feed pumps, &c., is taken from Tredgold to be equal to 0·075 S.H.P.

The horse-power due to the several elements worked out on the foregoing basis, when combined, form the *ship's horse-power* and may be tabulated as follows:

H. P. due to 1 =	E.H.P.	
„ „ „ 2 =	0·4 E.H.P.	
„ „ „ 3 =	0·1 E.H.P.	
„ „ „ 4 =		0·143 S.H.P.
„ „ „ 5 =		0·143 S.H.P.
„ „ „ 6 =		0·075 S.H.P.

$$\text{Total S.H.P.} = 1{\cdot}5 \text{ E.H.P.} + 0{\cdot}361 \text{ S.H.P.},$$
$$\text{or } 0{\cdot}639 \text{ S.H.P.} = 1{\cdot}5 \text{ E.H.P.}$$
$$\therefore \text{S.H.P.} = 2{\cdot}347 \text{ E.H.P.}$$

To this must be added the horse-power due to slip, which may be taken $=0{\cdot}1$ S.H.P., allowing for a slip of 10 per cent.

We then have:—

$$\text{I.H.P.} = 1{\cdot}1 \text{ S.H.P.}$$
$$= 2{\cdot}582 \text{ E.H.P.}$$
$$\text{or E.H.P.} = 0{\cdot}387 \text{ I.H.P.}$$

This agreed very fairly with the results of the actual experiments made. Mr. Froude's general experience pointed to the conclusion that at high speeds the effective horse-power was only about 37½ per cent. of the indicated horse-power; or that the indicated horse-power was 2·7 times the effective horse-power, which value he took as being more strictly reliable than that estimated above.

To reduce the above formula to such a form that it may be applicable to all speeds, it is necessary to keep the term involving the initial friction of the engines separate from the rest; for it represents simply a constant resistance operating at all speeds of the engine. Assuming, therefore, that the constant friction is equal throughout to 0·143 times, or one-seventh part of, the maximum load on the engines, we have the horse-power expended on friction for the maximum speed $= \frac{2\cdot7 \text{ E.H.P.}}{7} = 0\cdot385$ E.H.P., leaving the remainder, or 2·315 E.H.P., due to the other sources of expenditure, which are approximately proportional to the ship's true resistance.

To find the indicated horse-power due to the constant friction at any other speed of the engines, it is necessary to reduce the horse-power due to constant friction at the maximum power, in the direct ratio of the speed.

If V = maximum speed

v = any other speed;

Then the indicated horse-power due to the initial or constant friction at the speed v is

$$= \frac{v}{V} \times 0\cdot385 \text{ (E.H.P. due to V)},$$

and the total indicated horse-power corresponding to the speed v is,

$$\text{I.H.P.} = 2\cdot315 \text{ E.H.P.} + 0\cdot385 \frac{v}{V} \text{(E.H.P. due to V)};$$

or, separating the useful from the collateral expenditures of power,

$$\text{I.H.P.} = \text{E.H.P.} + 1\cdot315 \text{ E.H.P.} + 0\cdot385 \frac{v}{V} \text{(E.H.P. due to V)}.$$

The following example will illustrate the method of calculating and drawing these thrust curves. We will

take the case of the modified four-bladed screw-propeller of H.M.S. 'Iris,' of which the particulars are given in the table on page 412. From the four speeds there given, four points representing the indicated thrust can be obtained, and a fair curve must then be drawn through these points.

First draw the horizontal and vertical axes O X and O Y intersecting at O (Fig. 169). Divide O X into equal parts

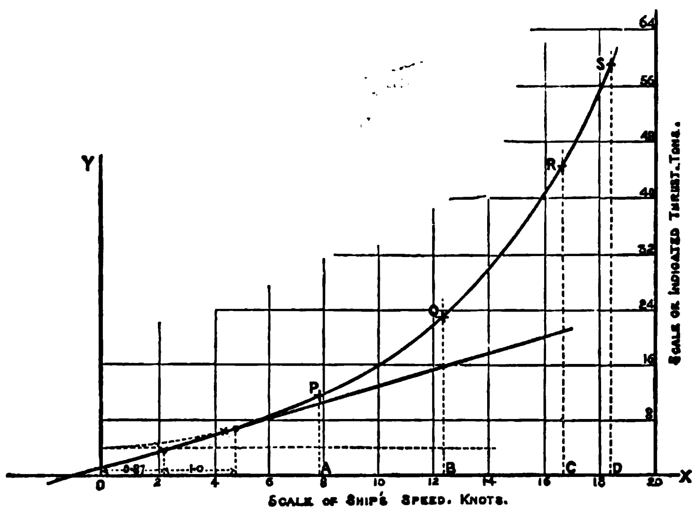


FIG. 169.

to represent knots per hour to a suitable scale, and divide the vertical ordinate to represent the thrusts in pounds, or tons, as may be the more convenient. From the points A, B, C, and D, representing the respective trial speeds of the ship, draw vertical ordinates.

$$\text{The indicated thrust} = \frac{\text{I. H. P.} \times 33{,}000}{\text{P} \times \text{N}}.$$

Therefore at the lowest speed of 7·797 knots,

$$\text{Indicated thrust} = \frac{606 \times 33{,}000}{19{,,}11\tfrac{1}{2} \times 40{\cdot}963}$$
$$= 24{,}460 \text{ pounds}$$
$$\text{or} = 10{\cdot}92 \text{ tons.}$$

On the ordinate drawn at A mark off A P $=10{\cdot}92$; then P will be one point on the curve.

Similarly for speed O B $=12{\cdot}279$ knots

$$\text{Indicated thrust} = \frac{1{,}833 \times 33{,}000}{19{,,}11\tfrac{1}{2} \times 61{\cdot}347}$$
$$= 49{,}400 \text{ pounds}$$
$$= 22{\cdot}05 \text{ tons.}$$

Then making B Q $=22{\cdot}05$, Q will be the second point in the curve.

For speed O C $=16{\cdot}564$ knots

$$\text{Indicated thrust} = \frac{5{,}108 \times 33{,}000}{19{,,}11\tfrac{1}{2} \times 85{\cdot}388}$$
$$= 96{,}900 \text{ pounds}$$
$$= 43{\cdot}26 \text{ tons.}$$

Making C R $=43{\cdot}26$, R will be the third point on the curve.

For speed O D $=18{\cdot}573$ knots

$$\text{Indicated thrust} = \frac{7{,}714 \times 33{,}000}{19{,,}11\tfrac{1}{2} \times 97{\cdot}189}$$
$$= 131{,}250 \text{ pounds}$$
$$= 58{\cdot}55 \text{ tons.}$$

Making then D S $=58{\cdot}55$, S will be the last point on the curve given by the trials. The fair curve drawn through these points P, Q, R, and S will be the indicated thrust curve for the ship. In order to determine the intersection of this curve with the axis O Y, which will represent the thrust due to the initial friction of the engines, we must proceed as directed on page 422.

It is very desirable, in order to insure accuracy in

determining the initial friction geometrically, that the ship should be tried at as low a speed as possible, say 3 to 4 knots an hour. It is, however, generally difficult to arrange for this in new ships, on account of the time and expense necessary, and the cases in which reliable data have been obtained at very low speeds are comparatively few; but it is clear that the lower the speeds at which the observations are taken, the more correctly can the equivalent of the constant friction be determined.

The application of these thrust curves, which are so easily and simply drawn, may be made by the engineer officers of the Navy the source of much valuable information with reference to the efficiency of performance of their machinery, without involving much extra trouble, and we feel sure that any time spent on them would be amply repaid by the results obtained. War-ships on ordinary service steam at various speeds, many of them very low, so that a series of points at the lower end of the curve, which are so difficult to get before the ships proceed on service, can be readily obtained. As before pointed out, this is the most important part of the curve, so far as the determination of the initial or constant friction of the engines is concerned. Indicator diagrams have, in accordance with the Admiralty instructions, to be taken at least daily, or oftener if any material variation of power occurs, so that in course of time a considerable number of diagrams are taken, showing the performance of the engines under a great variety of circumstances. Instead of the actual speed of the ship, which would generally be difficult to obtain with exactness, it will be sufficient for the required purpose to take the revolutions of the engines per minute, which is a fairly reliable measure of the speed of the ship under ordinary conditions. Of course it actually indicates the speed of the screw, which is equal to the pitch multiplied by the revolutions, and the values obtained

from the curves drawn by taking the revolutions of the engines to represent the speeds of the ship will give relative and not absolute values. The base line might therefore be divided to represent revolutions per minute, to a suitable scale, and for each number of revolutions per minute at which diagrams have been taken ordinates showing the indicated thrust, calculated as shown at p. 419, should be set up, and the fair curve drawn through the points thus obtained will approximately represent the thrust curve. In order to ascertain the initial or constant friction of the engines, the tangent should be drawn at the lowest point of the curve, and the abscissa, corresponding to the point at which the tangent is drawn, bisected, in order to determine the height corresponding to the initial friction, instead of being divided in the ratio of 0·87 to 1 as pointed out at p. 423. This will tend to correct the error introduced by taking the revolutions of the engines to represent the speed of the ship.

The curves thus constructed, though not absolutely exact, will give a very fair approximation to the actual efficiency of the ship, and if they be drawn systematically, under varying circumstances, there is little doubt that many useful results will follow. For example, if at any time the initial friction should be increased from any cause, such as the bearings getting out of line, packing rings at backs of slide-valves becoming inoperative, &c., it would be clearly shown on the diagram, by the point at which the curve would intersect the vertical axis.

The application of the curve in a few cases would suggest many other ways in which it might be made interesting and useful. For instance, if *coal curves* were combined with the *thrust curves*, by setting up ordinates representing the coal used per hour and per knot respectively, and drawing fair curves through them, many interesting results would probably be obtained. It is un-

necessary to suggest the many applications that might be made of this system, which will occur to the minds of all officers using it; and if it be applied intelligently and developed, there can be no doubt that much valuable information will be gained, and the duty of obtaining the most efficient and economical results from the machinery and boilers will be simplified. It is a usual custom for engineer officers to tabulate, for their own information, the average horse-power, coal, &c. required when the engines are working at various speeds; and we trust that this brief sketch, showing how the results may with very little trouble be shown graphically, may induce them to devote attention to it. The tabulated results only represent a number of isolated facts, but when they are shown in the form of a curve, as described, the general law underlying the facts can be more readily ascertained, and more valuable and important information obtained.

CHAPTER XXII.

PADDLE-WHEELS.

Radial paddle-wheel.—THE simplest form of paddle-wheel is generally known as the common or radial paddle-wheel. In this wheel the floats are bolted direct to the arms of the wheel, and consequently the pressure they produce on the water is always perpendicular to the radius. It is therefore clear that at any instant the only float that produces a direct sternward reaction is the one at the bottom of the wheel, all the others having a vertical component tending to raise or depress the vessel, which is entirely wasted so far as propulsion is concerned.

Width of floats.—The extreme width of the floats should not exceed one-half the width of the vessel, so that the combined width of the two paddle-wheels should not be greater than the width of the ship. In sea-going steamers the width of float generally does not exceed one-third the width of vessel. In still water, the greater the width of float the more effective the wheel, as the required area of race can be obtained with less immersion, and the loss from oblique action is thereby reduced. This condition, however, is limited by the practical difficulties involved in supporting the overhanging end of the paddle-shaft. In rough weather, extreme width would be objectionable from many causes.

Immersion of wheels.—The depth of immersion of paddle-wheels is practically limited by the draught of water of the vessel, as it is evidently undesirable to allow

the lower edge of the propeller to be below the keel. The immersion of the wheels must also depend on their diameter; for if the floats act too obliquely on entering and leaving the water, a large proportion of the power would be wasted in producing vertical reactions. As an extreme case we may point out that a radial paddle-wheel immersed to its centre would be of no value as a propeller.

In general the greatest immersion of a paddle-wheel should not exceed one-half the radius, or one-fourth the diameter, of the wheel. In sea-going steamers, for long voyages, the immersion at starting should be about one-half the radius, and the mean draught for the voyage about one-third the radius, of the wheel.

For effective working, the tops of the floats when in their lowest position should always be some distance below the surface of the water. In large sea-going steamers the top of the lowest float was usually about 18 to 20 inches below the surface at mean draught; in smaller vessels from 12 to 15 inches. In river steamers the immersion is generally much less, say from 3 to 6 inches; but these boats always work in smooth water and their draught is practically constant. In sea-going steamers the immersion of the floats at their lightest draught should not be less than 6 inches.

Number and pitch of floats.—In radial paddle-wheels the number of floats is generally made equal to the number of feet in the diameter of the wheel, which practically sets them at about 3 feet apart from each other. In some fast ships, to reduce vibration they have been set closer than this, or from 2 to 2½ feet apart. If the floats be set too closely together the water will not escape with sufficient freedom from between them, whilst if too far apart the vibratory action will be excessive. The number and pitch of floats should be so arranged that there

will always be at least three floats immersed at the same time.

Reefing paddle-wheels.—The floats are secured to the radial arms of the paddle-wheels by hook-bolts in such a manner that if the draught of the vessel be increased the floats may be readily unshipped and secured in other positions nearer the centre of the wheels. This operation is usually called *reefing the paddle-wheels*, and is obviously equivalent to reducing the effective diameter of the wheel and the immersion of the floats, and thereby diminishing the loss from oblique action. Reefing is desirable when by increased draught it is found that the wheels cannot be driven fast enough to utilise all the steam generated in the boilers. This operation, by decreasing the resistance, enables all the steam generated to be used, and the piston speed increased, with a consequent gain in the power and speed of the ship. Paddle-wheels should never be reefed to such an extent as to bring the top of the lowest float above the surface of the water, which would cause loss of thrust and power.

The efficiency of radial paddle-wheels is somewhat improved by making the floats in halves, the lower half being placed on the after and the upper half on the forward side of the radial arms.

The only points of advantage that can be adduced in favour of the radial over the feathering paddle-wheel are its lightness, simplicity, and cheapness of construction. There are no working parts in it, and in the event of injury, defects can be readily made good at little cost. Its propelling efficiency, however, is much less than that of the wheel with feathering floats, and the improvements in design and workmanship have made the latter so practically trustworthy, for the comparatively few services for which paddle-wheels are now required, that the radial

paddle-wheel may be regarded as altogether a propeller of the past.

Paddle-boxes.—The form of the paddle-boxes should be arranged to allow the water to pass freely away from the propellers, and not to be boxed up and carried round with the wheel, by which the thrust would be reduced and the loss of work increased.

Feathering paddle-wheel.—In order to obviate the disadvantages resulting from the oblique action of the floats of radial paddle-wheels, especially in cases where the draught of the vessel varied considerably, feathering paddle-wheels have been introduced. The general form and arrangement of these propellers are shown in Figs. 170 and 171. The wheel consists of a wrought-iron framework, secured to a strong cast-iron centre or boss, keyed on the end of the paddle-shaft. The floats, instead of being fixed to the arms of the wheel, are carried on joint-pins, and their motion is controlled by the action of an eccentric, through rods and levers, in such a manner as to keep the floats approximately normal to the effective surface during their passage through the water, so that the whole of the thrust will be in a sternward direction. Its efficiency is at least ten per cent. greater than that of the radial paddle-wheel when both work under suitable conditions, and the economy and efficiency resulting from its use far more than compensate for its increased first cost and expense of maintenance.

It will, however, be seen that its construction is somewhat complicated, and that it requires considerably more care and attention than the common radial wheel. It is very important that the working parts should be sufficiently strong to withstand the shocks to which they are exposed, without undue straining, for damage to any part of the feathering apparatus is liable to paralyse the action of the entire wheel. These wheels are consequently made

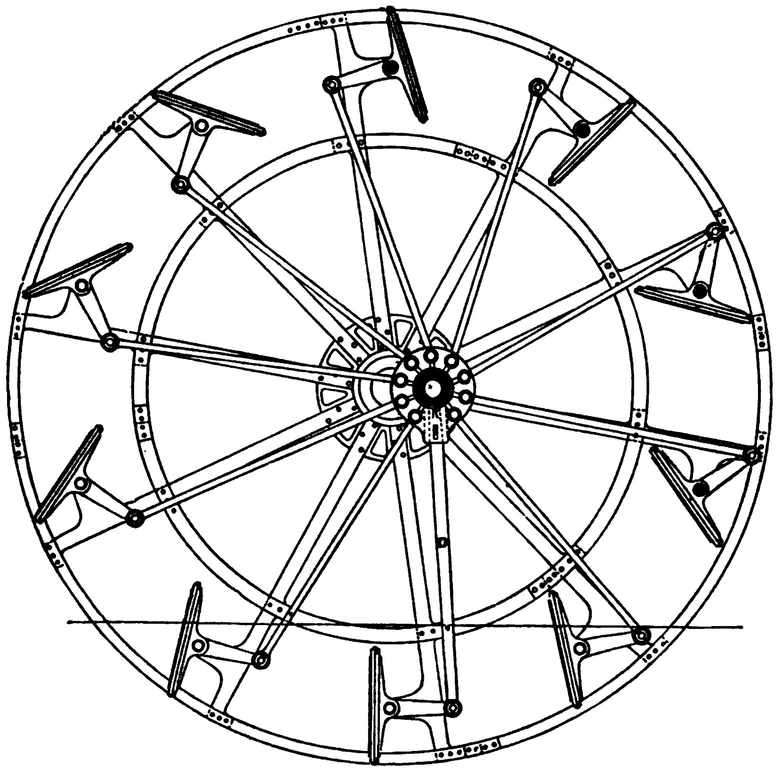


FIG. 170.

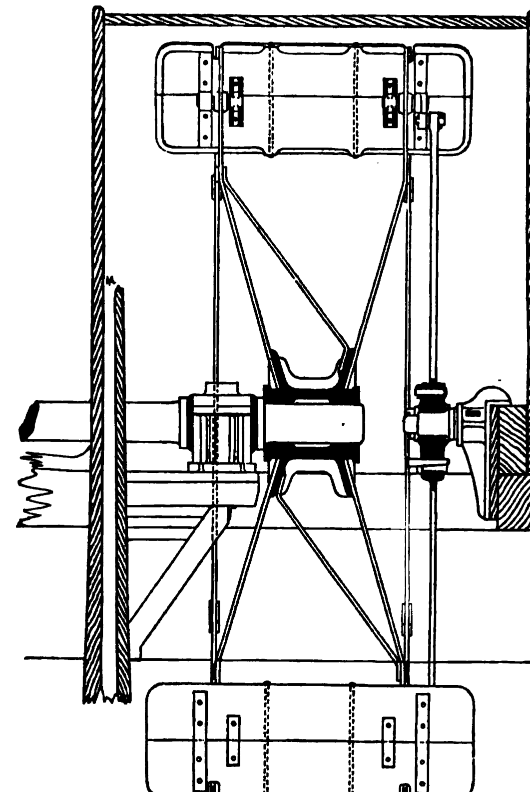


FIG. 171.

much heavier than the radial wheel, and are more difficult to properly support.

This complication and liability to serious injury might possibly have tended to prevent their being extensively used for long sea voyages, in preference to the simpler radial wheels, which, if damaged, could be so much more easily repaired. As, however, the paddle-wheel for ocean navigation has been entirely superseded by the screw-propeller, this point need not be further discussed, and there can be no doubt that for short voyages, river navigation, and towing purposes, for which alone paddle-wheels are now used, feathering floats possess very great advantages, enabling the wheels to be made of less diameter and width, and in consequence of their increased efficiency the indicated horse-power of the engines may be proportionately reduced for a given speed.

Dimensions and pitch of floats.—The floats in feathering paddle-wheels are generally placed about twice as far apart as the floats in the radial wheel; that is, the pitch of the floats is usually about six feet. They are also made deeper, say about twice the depth of the common float, for in this case the area of the race, or stream driven back on either side of the ship, is equal to the width multiplied by the depth of the float, instead of the width of float multiplied by the depth of immersion, as is assumed to be the case with the radial paddle-wheel.

Eccentricity of feathering apparatus.—The method of determining the throw and position of the eccentric necessary to produce the proper action of the floats in the water may be easily explained by means of a skeleton diagram. In Fig 172 let A represent the centre of the paddle-shaft, and K the centre of the eccentric pin or sheave that produces the necessary movement of the paddle-floats, the correct position of which is required to be found. For simplicity, the floats are supposed to be jointed at

their centres. In practice this is not exactly the case, the joint being just behind the float, and as close to it as possible. In an actual design, this would render necessary a slight modification in the details of the following method of determining the eccentricity, but the deviation is small, and there will be no difficulty in making the required correction when the principles involved are understood. The circle B C D E F G, drawn with A as centre, through the centres, or joints, of the floats, may be taken to

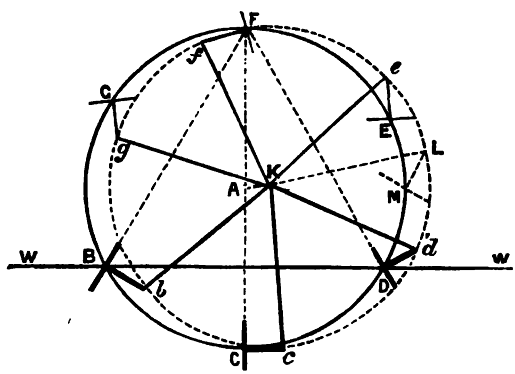


Fig. 172.

represent the paddle-wheel circle. Let W W represent the water-line, B and D being the points in which it is cut by the paddle-wheel circle.

Consider three floats in the positions shown by B, C, and D, one just entering the water, the second at its lowest point, and the third just leaving the water. In order that the motion of the floats through the water should be correct, moving as nearly as possible edgewise, relatively to the water in the paddle-race, the directions of the faces of these floats produced, should meet at the point F, at the top of the paddle-wheel circle. If, therefore, from F, the highest point of the circle, straight lines, F B, F C, and F D are drawn, these will represent the directions

of the faces of the paddle-floats at these respective points. From the centres of these three floats, B, C, and D, draw the float levers, B*b*, C*c*, D*d*. These are usually at right angles to the float, and their lengths are about three-fifths of the depth of float. These values are arbitrary, and subject to convenience in any particular design; but the angle seldom deviates much from a right angle, and the proportionate length of lever given above is generally suitable. Having thus determined the points, *b*, *c*, and *d*, to which the radius rods from the eccentric have to be jointed, it is only necessary to find by plane geometry the centre, K, of the circle passing through them. K will then be the centre, and A K the throw, of the eccentric necessary to produce the required motion of the floats.

The velocity of the propeller race is clearly represented by the circumferential velocity of the circle B D F G, and the effect of the motion produced by the action of the eccentric, thus determined, will be to cause the floats, while *in the water*, to move as nearly as possible edgewise, relatively to the propeller race, and thus prevent loss from oblique action. By drawing floats in other positions, it will be seen that their action when *out of the water* is far from being free from vertical reactions, but these, operating only on the air, may be practically neglected, as they do not affect the propulsion of the ship, which is only influenced by the action of the floats when in the water.

Maximum depth of float.—In order to determine whether the float will clear the radius rod for the complete revolution, or if it will require to be notched or otherwise modified, it will be necessary to draw a float in position when the radius rod is in the line joining the two centres A and K, in which position the angle will be greatest. Produce A K, to cut the dotted circle drawn

through the joints of the radius rods at the point L. From L, lay off L M equal to the length of the float-lever to cut the paddle-wheel circle, B D G, at the point M. Then M will represent the centre or joint of the float. If the float be drawn at M perpendicular to the lever L M, it will be seen what modification, if any, is necessary; for this will give the relative positions of the respective parts when the radius or guide-rod approaches most closely to the inner edge of the paddle-float.

Paddle-shaft bearings.—The shaft carrying the paddle-wheel is called the paddle-shaft, and is sometimes supported by two bearings, one on the ship's side, and the other on a beam, called the *sponson or spring beam*, on the outside of the paddle-box. In this case, the feathering apparatus has to be worked by a large eccentric on the paddle-shaft, to which the radius rods are attached.

Overhung wheels.—The most general arrangement, however, is that shown in Figs. 170 and 171, in which the paddle-wheel is overhung and supported by a single bearing on the ship's side, the outer bearing being dispensed with. In this case the feathering motion is produced by attaching the radius rods to a sheave working on a pin carried by a bracket fixed to the outer side of the paddle-box, in the proper position, eccentric to the wheel, to produce the required movements of the floats.

Driving and radius rods.—In the feathering apparatus, one of the guide or radius rods, called the driving rod, is rigidly fixed to the eccentric, to make it rotate about the axis K. The remainder of the rods are simply jointed to the eccentric, as well as to the float-levers, with pins. In Fig. 170 the driving rod is marked D. All the joints in the feathering apparatus should be bushed either with brass, white-metal, or lignum-vitæ.

Paddle-shaft bearings.—It is clear that the outer bearings of the paddle-wheels, when they are so fitted, cannot

be examined when the engines are at work. Guide-boards or troughs are therefore fitted on the side of the paddle-box, so that the water carried up by the wheel is caused to constantly run on these bearings to prevent their overheating. This splashing and churning action of the wheel on the water is also often utilised for the purpose of keeping a small tank, fitted inside the paddle-box, always full of water, to be used, if necessary, on the bearings of the paddle and intermediate shafts, which are above the water-line. The water-service pipes for these journals are also, in general, connected with the delivery pipe from one of the auxiliary pumping engines of the ship. When the paddle-wheels are overhung, and carried by a single bearing on the ship's side, the journal should be made of larger diameter and considerably longer than is necessary when an outside bearing is fitted, to resist the increased pressure and strains. There should also be thrust collars on the journal, to prevent end motion when the ship rolls. The bearings for paddle-shafts in the Royal Navy are generally made of brass, though they are sometimes made of lignum-vitæ strips, as in the case of bearings for screw-shafts. When so fitted the shafts should be cased with brass.

Stuffing-box on ship's side.—The hole in the ship's side through which the paddle-shaft passes is either fitted with an ordinary stuffing-box, or covered with a leather disc to prevent the passage into the ship of water carried round with the paddle-wheel.

Disconnecting apparatus.—In paddle-wheel tugboats, gear is usually fitted to enable the wheels to be disconnected from each other, and each engine worked independently, to facilitate the manœuvring of the vessel. In many cases an ordinary disconnecting clutch is fitted on the intermediate shaft for this purpose. Another plan for disconnecting the engines from each other is shown in

Fig. 173. A cast-iron disc, D, is fitted on the intermediate shaft, in lieu of a crank-arm. This is driven by feathers on the shaft, over which it may be drawn back, clear from the crank-pin, when the engines are required to be worked independently. With this arrangement it is desirable that the paddle-shaft should be fitted with an outer

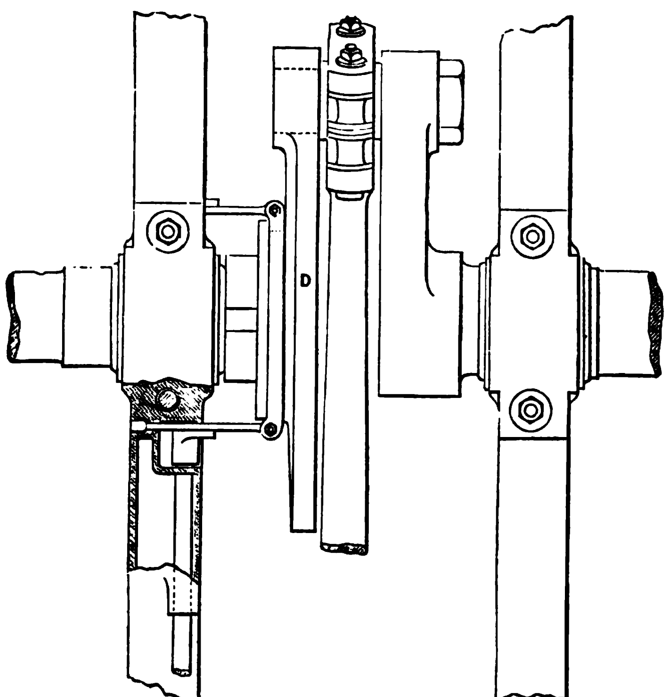


FIG. 173.

bearing: for otherwise the shaft, when disconnected, would only be working in a single bearing, which would cause severe bending strains. Engines of this class, of large power, should be fitted with auxiliary slide-valves to further facilitate handling, as the main slide-valves are heavy and difficult to work by hand.

In the more recent paddle-wheel tugboats in her Majesty's service a pair of cylinders has been attached to each crank. The shafts for each wheel in this case may either be connected by a clutch or friction coupling, or left quite independent of each other, for it is clear that the engines will be entirely under control whether they are coupled or not.

In the earlier paddle-wheel vessels in the Royal Navy, arrangements were fitted to enable the wheels, with their

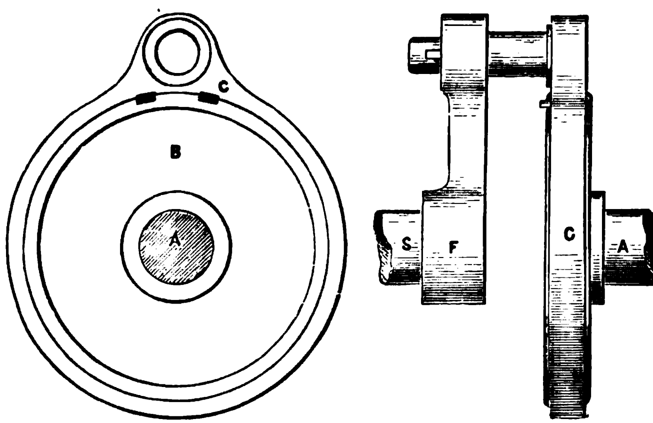


FIG. 174.

shafts, to be disconnected from the intermediate shafts when desired, so that the ships might proceed under sail alone without the engines being caused to revolve by the action of the paddle-wheels. One method of effecting this is shown in Fig. 174. A cast-iron circular disc or sheave B is keyed on the inner end of the paddle-shaft A. A wrought-iron strap, C, fits over the periphery of this disc, with a projection at one part, in which a hole is bored to receive the end of the crank-pin. Keys are fitted by means of which the strap and disc may be secured together to transmit the power of the engines to

the wheels; S is the end of the intermediate shaft, and F the crank-arm. When the ship was under sail alone, the keys were withdrawn, and the disc then revolved with the paddle-wheels and shafts without transmitting the motion to the intermediate shaft and engines. These fittings are, however, now obsolete, as no paddle steamers are built which are supposed to be efficient under sail alone, their use being confined to short voyages and river work, and they are essentially and solely steamers.

CHAPTER XXVII.

SCREW-PROPELLERS.

EACH blade of a screw-propeller may be regarded as a small portion of the thread of a screw of great pitch, and of considerable depth relatively to the pitch. The generation of the surface of a propeller blade of uniform pitch may be conceived from the following geometrical construction.

Let A A′, Fig. 175, represent the axis of the screw.

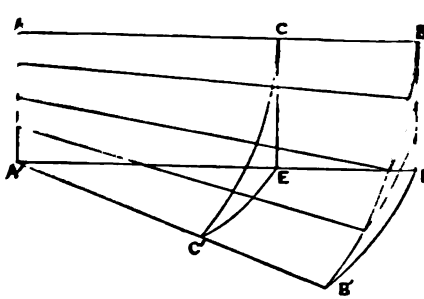


FIG. 175.

Suppose a line A B, perpendicular to A A′, to move uniformly along A A′, and at the same time to revolve uniformly around it. It is clear that the extremity B of the arm A B, will travel on the surface of a cylinder, and will, in point of fact, trace out a spiral curve on that cylinder. The same will be true of any and every point in the line A B,

and the surface swept out or developed by A B will be a spiral or screw surface.

Pitch.—If the line A B made a complete revolution around A A′, the distance of A′ from A at the end of the revolution would be the pitch of the screw. It is obviously equal to the distance between two consecutive threads measured in the direction of the axis.

Length of screw.—An actual screw-blade only consists of a portion of a complete convolution; and the extreme dimension of the blade, measured in the direction of the axis, is called the *length of the screw*. In Fig. 175 this is represented by the line A A′. The length of the screw may also be defined as the greatest length of the cylinder described by the blades in revolving; and the aggregate length of all the screw-blades is equal to the length of the screw multiplied by the number of blades.

Angle of the screw.—The angle B B′ D between the curve and the plane A′ D B′ perpendicular to the axis, is called the *angle of the screw*, at the radius A B. It is evident that the smaller the radius the greater will be the angle of the screw, the angle C C′ E, for example, being considerably greater than the angle B B′ D. In propeller-blades having constant pitch throughout, the angle of the blade increases the nearer the centre is approached.

By considering the method of generation of a screw surface it will be seen that the relations between the pitch, circumference, and angle of the screw may be shown by means of a right-angled triangle, having the pitch as perpendicular and the circumference as base, the tangent of the angle of the screw being equal to the pitch divided by the circumference. In Fig. 176, let A B represent the pitch and B C the circumference of a screw, to any given scale. Then the angle A C B will represent the angle of the screw at the end of the blade. The pitch being assumed the same throughout the blade, the angle for any

other part of the blade may be found by determining the circumference B D at the given part, and joining A D; the angle A D B being the required angle.

Form of blade.—Screw-blades are made of a great variety of forms, and we shall only be able to illustrate a few of the more general. The effect of form of blade has not yet been fully ascertained, but it would appear that the shape suitable for one ship does not always prove equally efficient for another ship, and in our present state of knowledge of the subject it is difficult to predict with certainty the form of blade that will be most suitable for any particular ship. As a rule it would appear that form has no peculiar value as regards propulsive efficiency, though it may have some influence on the amount of vibration

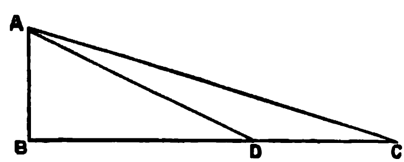


FIG. 176.

produced. There can, we think, be little doubt that the main points to be considered are, the pitch and surface of the blades. The work absorbed in friction and the disturbing effect on the stream-line motions must necessarily have some effect in determining the most suitable shape of the blade, but this has not yet been reduced to exact calculation.

In consequence of the increased angle of the screw-blade as it approaches the axis, the inner part of the blade, if the boss be small, has very little propulsive efficiency, and only absorbs power in churning and agitating the water. This was very noticeable in the earlier forms of screw-propellers, in which the central part, or boss, was of very small diameter, say about twice the diameter of the

shaft, so that the inner portions of the blades were nearly in a fore and aft direction. In these screws also, the length of the screw, or, in other words, the length of the projection of the blades on a fore and aft plane, was constant throughout the blades, so that the side view of the screw was rectangular. The ends of the blades were, therefore, very broad, and absorbed much power in surface friction.

Griffiths' screw.—To obviate this, Mr. Robert Griffiths substituted for the central inefficient portion of the screw a large spherical boss, one-third the whole diameter of the propeller, which would revolve without agitating the water. This principle is now generally adopted for all screw-propellers, though the bosses are not usually quite so large as in the earlier Griffiths' screws. The general diameter of the boss in screw-propellers as now fitted is about one-fifth to one-quarter the diameter of the propeller. In the Griffiths' screw also, the outer ends of the blades, which revolve at the highest velocity, were considerably shortened, to reduce loss from friction. The widest part of the blade was about four-tenths of the radius from the centre, the blade being somewhat pear-shaped. The tips of the blades were bent forward towards the ship, to the extent of about one-twenty-fourth of the diameter of the propeller. The rounding of the blades towards the tip probably renders the screw less liable to foul.

The following may be taken as an example of the proportions of the Griffiths' propeller relatively to the pitch:

Width of blade at tip	0·07	pitch
Greatest width	0·167	„
Width at root	0·11	„
Aggregate length of two blades	0·24	„

The two-bladed propeller shown in Fig. 184 is an example of the modified form of Griffiths' screw as fitted in the more recent ships.

The usual numbers of blades in screw-propellers are two, three, and four. It will suffice for our present purpose to confine attention to two- and four-bladed screws, which are the most general in the Royal Navy, the former being necessary in cases where it is intended to lift the propeller out of the water when the ship is under sail in order to diminish the resistance.

The pitch of the blades is sometimes uniform, but frequently the pitch of the leading half of the blade, or the part that first strikes the water, is made less than that of the following half, to decrease shock in the action of the blade on the water.

In the mercantile marine, screw-propellers are usually made of cast-iron, the blades being cast on the boss so that the whole propeller forms a single casting. In the Royal Navy the screws are generally made of gun-metal, and the blades are cast separately from the boss, to which they are secured by strong gun-metal bolts, the holes in the flanges of the blades being elongated to allow the pitch of the screw to be adjusted, within certain limits, when necessary.

Screw-propellers can be made much thinner and lighter of gun-metal than of cast-iron, and by making the blades separate from the boss the whole screw is not destroyed in the event of damage to a single blade. Spare propeller-blades are carried in all ships of the Royal Navy, to be fitted in case of accident. In some ships, cast-steel has been used for screw-propellers. By the employment of this material the blades can be made thinner, so as to reduce the edgewise resistance, and as the processes of manufacture become further developed, and sound castings of large size produced at reasonable cost insured, its application for this purpose will probably be extended. The backs of cast-steel blades have been found to suffer considerably from corrosion, and no efficient remedy for this has yet been devised.

Alteration of pitch.—The alteration of the pitch of the propeller is desirable when the working pressure of steam in the boilers is reduced as they become worn. The *weight* of steam that can be produced by the boilers is practically constant at all pressures; consequently, at the reduced pressures the *volume* of the steam generated will be greater than at the original pressure. In order to utilise all the steam at the lower pressures, it is generally necessary to drive the engines at a higher number of revolutions per minute; and to do this the pitch of the screw requires to be reduced accordingly. This operation of decreasing the pitch of the screw is therefore nearly always performed, in the ships of the Royal Navy, when the load on the safety-valves is reduced to any considerable extent.

For general work the four-bladed screw propeller is probably more efficient than the two-bladed propeller; it certainly has the effect of reducing *vibration*, which is principally due to the shocks produced by the water carried round with the propeller striking against the stern-post, and it is clear that the greater the number of blades the more continuous and less vibratory will this action be. If the blades were infinitely thin, it is probable that the screw with the most numerous blades would be the most efficient, but as the blades must necessarily be of considerable thickness to give them adequate strength, the work absorbed in forcing them through the water tends to balance their greater theoretical efficiency, and there appears to be no advantage in increasing the number of blades beyond four.

The acting surface of the screw-propeller, or its after face, preserves the exact geometrical form. The form of the forward face is modified by the thickness that must be given to the blade to provide the necessary strength. The horizontal sections of the blades approximate in shape to

semi-ellipses, shallow at the ends and becoming fuller towards the root. The surfaces of the blades should be made as smooth as possible to reduce the coefficient of friction.

Strength of propeller-blades.—The propeller-blade should be at least strong enough to withstand its share of the twisting moment on the shaft, but it should be weaker than the shaft with reference to the straining action of a blow from a hard substance on the tip of the blade. The moment of resistance of the blade in going from the boss to the tip should also diminish faster than the moment of such a blow, so that if fracture resulted it should take place as near to the tip of the blade as possible, so that the broken blade may be still useful for propulsion.

Professor Rankine gives the following rule for the thickness of gun-metal blades of ordinary section, in which the working strength may be taken from 4,000 to 4,500 pounds per square inch. The working strength of the shaft is taken from 8,000 to 9,000 pounds per square inch.

Rule :—Multiply the cube of the diameter of the shaft by four and divide by the number of blades, and by the length of the blade at its root measured parallel to the axis: the square root of the quotient will be the greatest thickness of a blade *at the axis of the shaft*, supposing the blade to be continued inwards so far :

If d = diameter of shaft,
N = No. of blades,
L = length of blade at root,

$$\text{thickness at axis} = \sqrt{\frac{4\,d^3}{N \times L}}.$$

If the blades are made of material having equal strength with the shaft, the multiplier will be 2 instead of 4.

In Fig. 177, let A represent the axis of the shaft, the circle G D H the outline of the boss of the propeller, and

A B a radius drawn from centre of shaft to tip of blade. Set off A C perpendicular to A B, to represent the thickness of blade as calculated by the foregoing rule, and join C B. Then the triangle B A C will represent the section through the centre of a blade that fulfils the condition of gradually diminishing in strength from the centre to the tip. In practice the end of the blade cannot be reduced to a point as shown; so from B, a distance B E must be set off equal to the minimum practicable thickness, and a line E F drawn parallel to the face A B. The point F will then be the weakest point of the blade, at which it will probably give way if fracture should result from the blow of a hard body on the tip when the engines are at work.

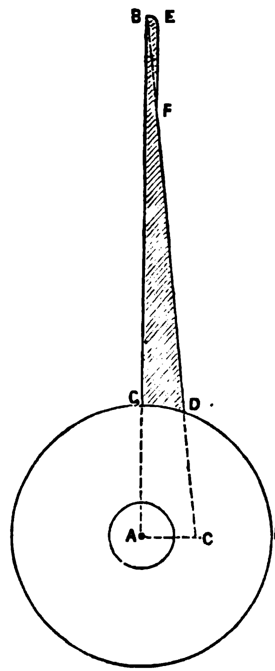


FIG. 177.

Aggregate length of the screw-blades.—The aggregate length of the screw-blades is kept practically constant whatever the number of blades may be; that is to say, the total surface of a screw-propeller of given diameter and power would be about the same whether it was divided between two, three, or four blades. The most advantageous aggregate length of blades is from 0·25 to 0·45 of the pitch. For two-bladed propellers this would give a length of screw of from one-eighth to one-quarter of the pitch, and in a four-bladed propeller from one-sixteenth to one-eighth of the pitch. As a rule the lower values are taken, as the work expended on surface friction is thereby

reduced. With more than four blades the aggregate length requires to be increased to some extent.

Four-bladed propeller.—Figs. 178 and 179 show an ordinary form of four-bladed screw-propeller. The blades are bolted to a strong boss secured on the end of the screw-shaft, the method of connection being clearly shown in the diagrams. The hole through the boss is tapered, and the screw is driven by the action of two longitudinal keys or feathers let into the shaft, and fitting into suitable key-ways cut in the boss. The end of the shaft is screwed, and the propeller boss is kept in its place by means of a nut, as shown in Fig. 179. A cap is jointed around the nut, to prevent the corrosive action of the water on the end of the shaft. The gun-metal liner on the stern shaft is tightly recessed into the boss for some distance, to prevent the access of water to the shaft at its junction with the propeller. The holes in the flanges of the blades are elongated, as shown in Fig. 179, to enable the pitch of the screw to be adjusted as may be required. When the pitch is set, brass or lignum-vitæ stops are fitted between the bolts and the edges of the holes, to prevent the pitch shifting. To insure a rigid connection, the boss is carefully fitted to the shaft, and is then slightly warmed and forced on by hydraulic pressure, before the nut is screwed up tight.

Hirsch screw-propeller. — Figs. 180 and 181 show the end and side views of a four-bladed Hirsch screw-propeller. In this propeller the blades are curved forward in a somewhat peculiar manner, so that the point of the blade is the first to act on the water instead of the whole edge of the blade, as in an ordinary screw. The direction of rotation is shown by the arrows. The axis of the blade is approximately a spiral curve, instead of being a straight line, and there are other minor differences in detail. The method of obtaining the curve of the centre

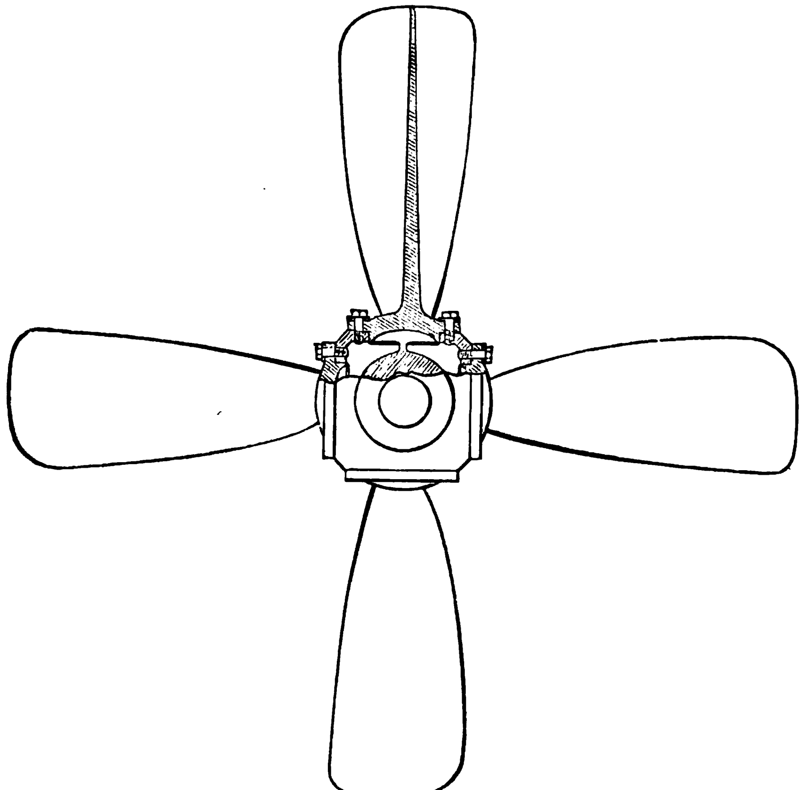


Fig. 178.

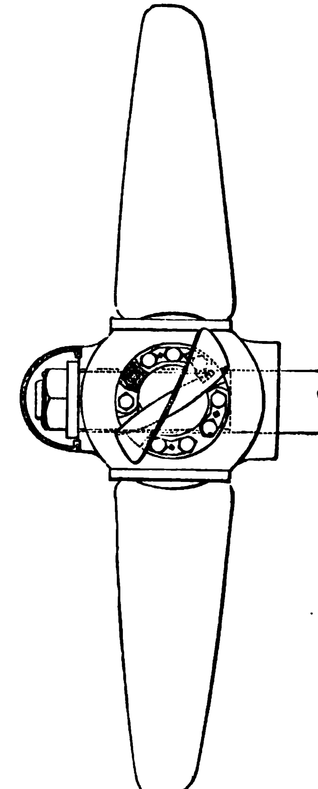


Fig. 179.

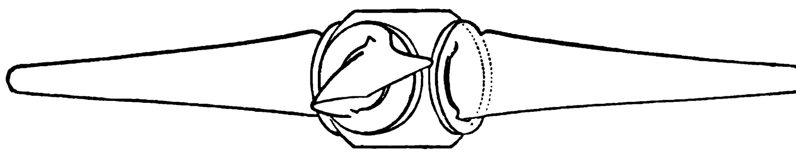


Fig. 181.

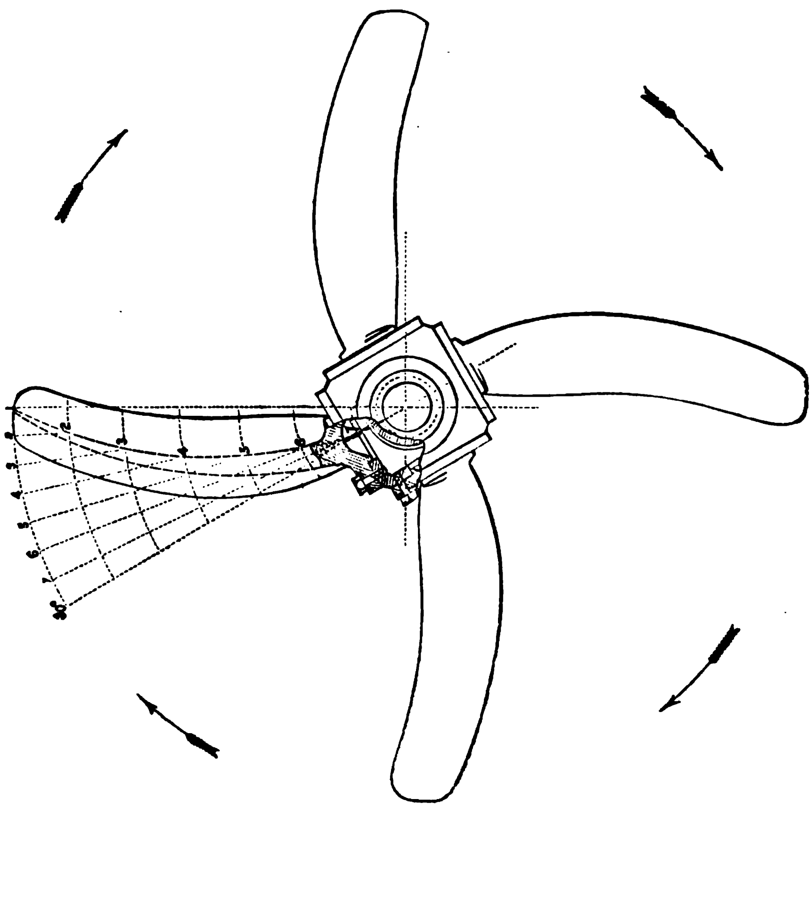


Fig. 180.

line of the blade is shown by the dotted construction lines on the top blade in Fig. 180.

It is supposed that this curved form of blade tends to resist the centrifugal motion of the particles of water acted on by the propeller, and to cause them to move directly sternwards, and also that shock, and the consequent vibration, is diminished by the action of the blade on the water being to a certain extent gradual, commencing at the point and passing onwards to the root. In the outer halves of the blades the sections have an increasing pitch, but nearer the root, where the angle is greater, the following pitch is somewhat less than that of the leading half of the blade, to let the water escape freely, instead of being carried round with the propeller. The variation in pitch of the several sections is so arranged that the mean pitch of the blade is equal to the pitch of the leading half of the blade. A large number of vessels have been fitted with these propellers, and from the published results it would appear that vibration has been considerably reduced and the propulsive efficiency somewhat increased. It is, however, possible that they are not so efficient as ordinary screws for working astern, and are more liable to foul. When fitted for lifting, a wider screw-well is necessary than for common screws, in consequence of the form of blade, and it is probable that they will be found to be most efficient and useful as four-bladed propellers, in which form they have generally given good results.

Disconnecting coupling.—When four-bladed screws are fitted to ships that are intended at times to proceed under sail alone, disconnecting couplings are fitted on the screw-shafting, so that the propeller may be disengaged from the engines and allowed to revolve freely without moving the engines. This is effected by fitting one of the couplings with bolts, or driving pins, that can be readily withdrawn by means of screws when required. A friction strap is

fitted over the after side of the coupling, to hold the propeller during the time the bolts are being withdrawn or replaced. A sketch of an arrangement of this kind is shown in Fig. 182. The small bolts marked B are fitted for the purpose of preventing the shafting being forced forward by the friction of the driving-bolts. It is necessary to fit a thrust-bearing abaft the disconnecting coupling,

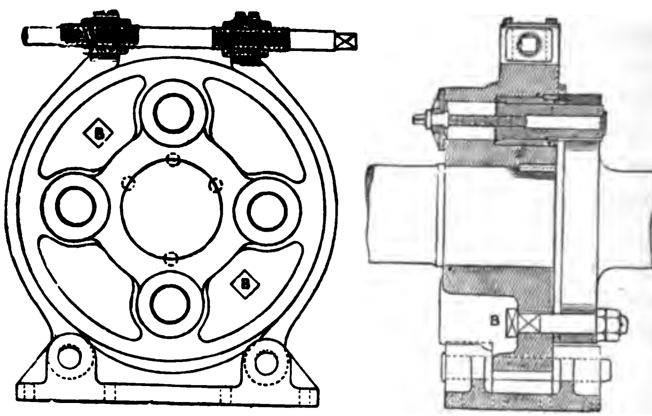


FIG. 182.

to prevent the shaft being drawn back by the action of the propeller when revolving disconnected.

Friction coupling.—In some cases the power of the engine has been transmitted from the crank-shaft to the screw-shaft by means of a conical friction coupling, as shown in Fig. 183. This system permits the ready disconnection of the propeller from the engines, but it is doubtful if it is as safe and reliable as the plan shown in Fig. 182, and it has not been so generally adopted.

With four-bladed screws, the increased resistance when the ship is under sail is less when they are permitted to revolve freely than when they are fixed; but it is not clear that this is the case with two-bladed screws

fixed in a vertical position, in which the blades are to a certain extent masked by the deadwood. It is probable that in this case there would be little difference in the resistance, whether the screw were allowed to revolve freely or fixed with the blades in a vertical position. Disconnecting apparatus would, however, be equally necessary in this

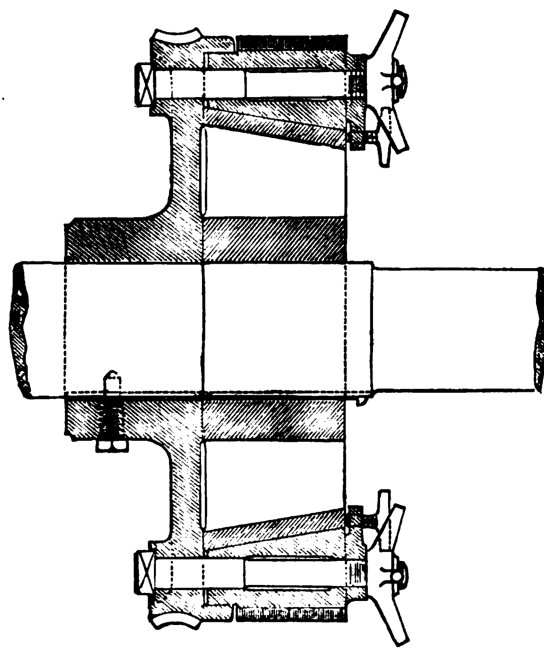


FIG. 183.

case to allow the engines to be turned by hand, when the screw was fixed, for the purpose of examining the engines and keeping them in good working order.

The increased resistance offered by the screw-propeller when the ship is under sail is, however, always an important item, and in order to obviate this, the following plans have been adopted in various instances.

1. To lift the screw entirely out of the water.

2. To arrange the blades of the screw so that they may be set approximately in line with the ship, or, as it is technically called, *feathered.*

3. The fitting of Mangin screws, which have two or more narrow blades on each side of the boss, behind each other, to reduce their transverse width and enable them to be more completely masked by the stern-post.

Lifting screws.—The earlier single-screw ships in the Royal Navy had their screw-propellers constructed and fitted to be lifted entirely out of the water when the ships were under sail, so that they should not impede the progress. A screw-propeller of this description, with the necessary fittings, is shown in Fig. 184. On the end of the screw-shaft a gun-metal coupling is keyed, having a rectangular slot running across it. This is generally called the *cheese-coupling.* The boss of the screw is cast with journals on its forward and after sides. The foremost journal has a T-head which fits into the slot in the cheese-coupling, by which the rotation of the screw-shafting is transmitted to the propeller when it is in place. A plan of the cheese-coupling and T-head for a lifting propeller is shown in Fig. 185.

Banjo-frame.—The journals on either side of the propeller boss of a lifting propeller are carried by bearings or plummer-blocks, in the ends of a rectangular frame, which is usually known as the banjo-frame. The bearings are fitted with strips of lignum-vitæ. When lowered into the working position, these bearings drop into brackets, technically called 'chairs,' securely bolted to the stern- and rudder-posts of the ship. Spigots or pintles are cast on the caps, or lower brasses, of the banjo-frame bearings, which fit into holes in the bottoms of the chairs. End views of these chairs, and banjo-frame bearings, are shown in Figs. 186 and 187, with the lignum-vitæ pieces for

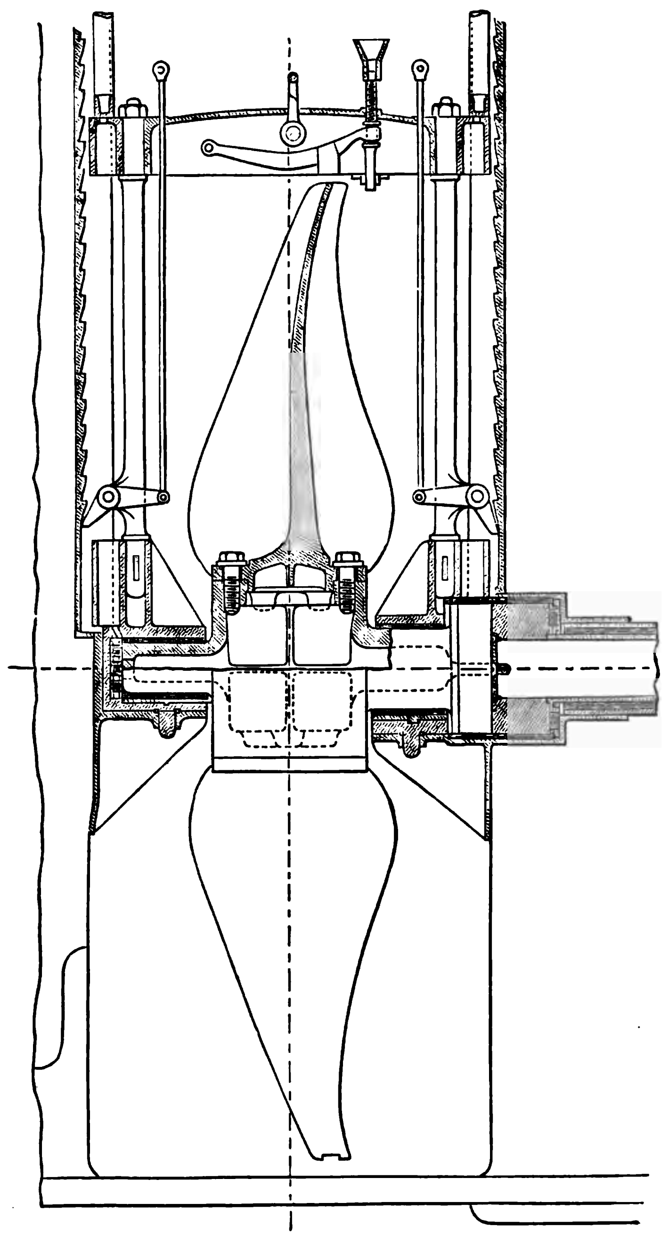


Fig. 184.

taking the forward and backward thrusts respectively; Fig. 186 showing the chair on the stern-post, and Fig. 187 the chair on the rudder-post. The banjo-frame works

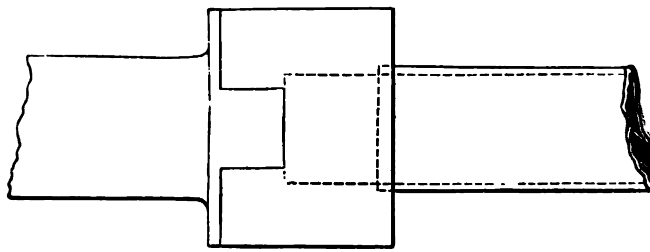


Fig. 185.

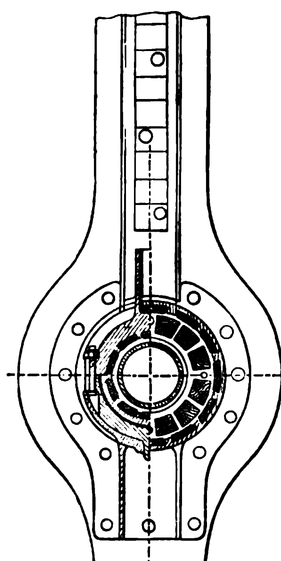


Fig. 186.

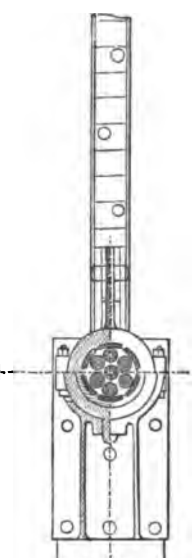


Fig. 187.

in guides on the stern- and rudder-posts of the ship, and in these guides are ratchet teeth, into which safety catches or pawls, attached to the banjo-frame, drop, to hold the

propeller in case of accident to the lifting gear while the screw is being raised. These are shown in Fig. 184.

Lifting gear.—For small screws, ordinary pulley-blocks hooked on to a shackle on the top of the banjo-frame are sufficient for lifting. For large screw-propellers, sheaves are fitted to the top of the banjo-frame, and rope falls passed round them and over corresponding pulleys fixed above the screw-well, are led away to the ship's capstan, or to tackles and gear worked on deck.

Lifting sheers.—In ships that have no poop, it is necessary to erect sheer-legs over the screw-well to carry the pulleys, in order to enable the screw-propeller to be lifted high enough to be clear of the water.

Lifting girder or strongback.—In ships fitted with poops, sufficient height can generally be obtained without the use of sheer-legs, and in these a permanent lifting girder fitted with sheaves, sometimes called a strongback, is fixed over the top of the screw-well. This is shown in Fig. 188, which gives a section through the stern of a ship, showing the screw-well with the propeller both in the working and raised positions, with all the necessary fittings for securing the propeller when at work and for lifting it out of the water when the ship is under sail alone.

Locking gear.—A lever stop is fitted on the top of the banjo-frame which can be screwed into a corresponding slot or notch in the top of the propeller-blade, to keep it fixed in the vertical position during the operation of lifting the propeller. The screw-shaft should be marked to show when the propeller is in the proper position for lifting.

Samson posts.—The banjo-frame and propeller are secured in their working position in the chairs by the action of struts, called technically the Samson posts, which abut against suitable brackets at the top of the screw-well, as shown in Fig. 188, in which the Samson

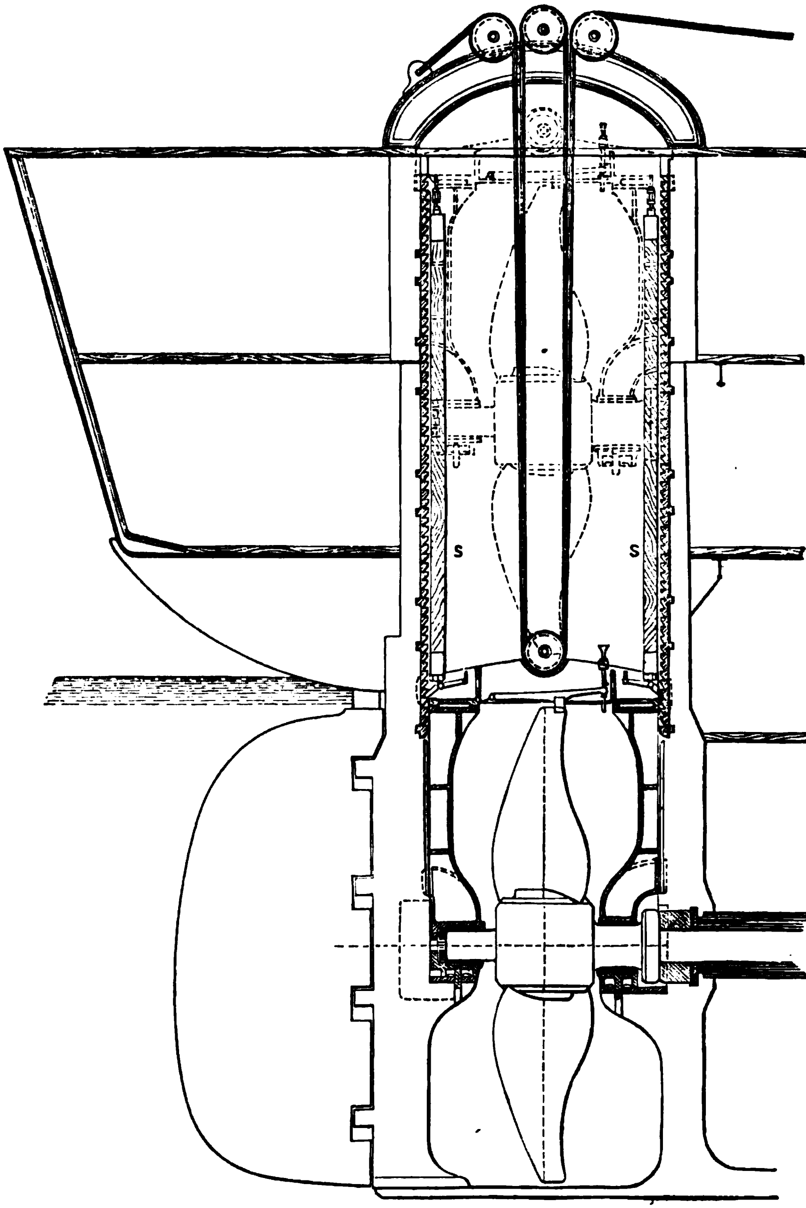


Fig. 188.

posts are marked s. Fittings must be provided to prevent these struts from becoming slack, and also for preventing the locking lever dropping to touch the blades, when the propeller is at work.

Propeller-supporting gear. — The propeller, when raised for sailing, is supported by means of chains or struts under the top of the banjo-frame, so as to relieve the strain from the pawls, and allow the lifting ropes to be removed. In some small ships bars are placed underneath the banjo-frame, when in its raised position, to support the weight.

In order to enable the bearings at the ends of the banjo-frame to be examined, and removed if necessary, without entirely lifting the screw out of the well, which would be inconvenient and often impossible in large ships at sea, additional brackets and gear are supplied, which can be attached to the screw boss or blade, and enable it to be supported independently of the banjo-frame. When this gear is fixed, the banjo-frame is free, and the bearings can be examined and re-lined with lignum-vitæ, or spare brasses fitted, as may be necessary.

There are many practical objections to the application of lifting screws, the most important being that they necessitate large apertures or screw-wells being built in the sterns of the ships, which not only occupy valuable space that might be used for cabin accommodation or stern guns, but also seriously interfere with the structural arrangements and strength. There is also considerable wear on the driving faces of the couplings and on the bearings, which produces backlash and unsatisfactory working, and entails much expense in repairs. The strains on the Samson posts, when the engines are working at full power, are great, and any accident to them, or even slacking back of the securing screws, would be attended with serious results. When the screw is lifted, if any foreign

substance should lodge on the chairs, it will be impossible to get the propeller back again into its working position without the aid of a diver, and in cases where the obstacle is not sufficiently thick to call particular attention to it in fixing the Samson posts, it is probable that the screw-propeller may be secured out of line with the shaft, causing incorrect working and increased wear.

There are many other minor inconveniences attending the use of lifting screw-propellers, so that attention has been turned to the design of arrangements by means of which the ship may be enabled to proceed under sail alone without undue increased resistance from the action of the propeller on the water, when it is fixed in a vertical position. The most general plans adopted for this purpose are screws with feathering blades, and the Mangin screw-propeller, which we now proceed to describe.

Feathering screw-propeller.—In this plan of propeller the blades are specially fitted to be turned in an almost fore-and-aft direction when the ship is under sail, so that they produce but little retarding effect on the vessel. There are several methods of effecting this operation. One of the neatest and most practical is that patented by Mr. R. R. Bevis, of Birkenhead. This arrangement is shown in Figs. 189 and 190. A short shaft or spindle, at the root of each blade, passes inside the hollow boss, and has a lever or arm attached firmly to it. The stern-shaft is made hollow, and rods are carried, through the centre of the shaft, from the ends of the arms on the blade-spindles to a collar near the forward end of the shaft, which can be moved forward or backward by means of a nut working over a thread cut on the outside of the shaft. The details are clearly shown in the diagrams. The dotted lines in Fig. 190 show the positions of the arms when the blades are set at the working angle, and the full lines their positions when the blades are feathered and placed in a fore-and-aft

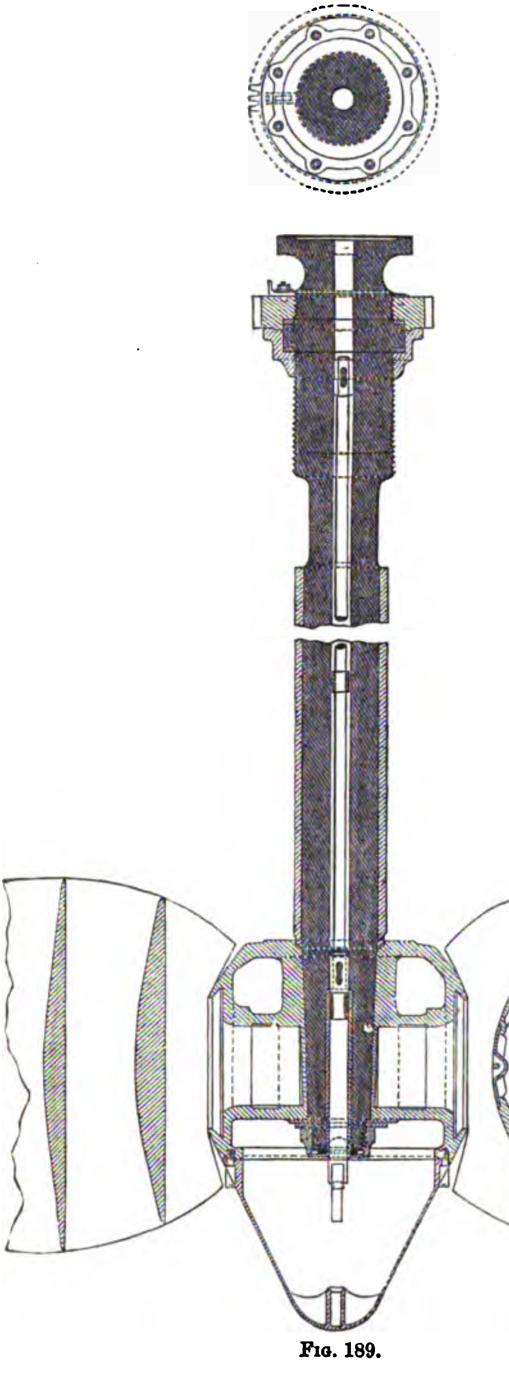


Fig. 189.

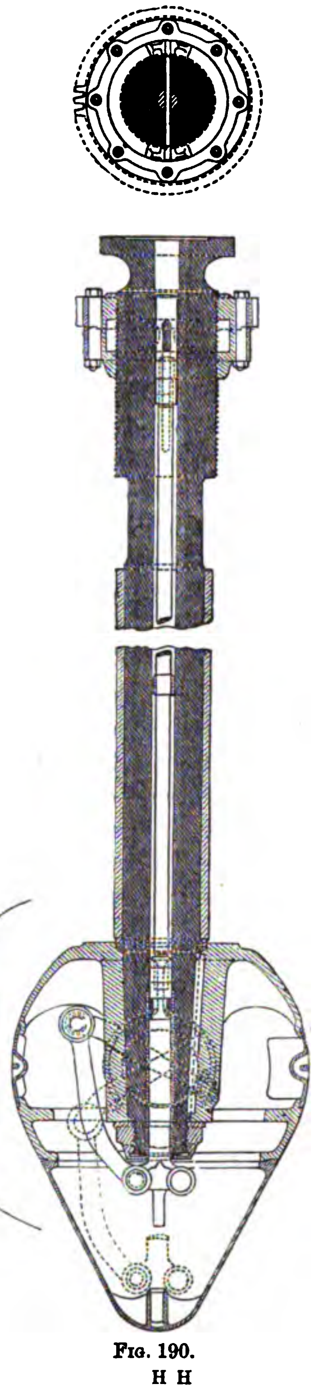


Fig. 190.

H H

direction. These propellers have been fitted to a large number of steam yachts, and to several ships in the Royal Navy, including steel corvettes of the 'Calliope' class, the engines of which are capable of being worked up to 4,000 indicated horse-power. This arrangement has the additional advantage of enabling the propeller-blades to be readily set to any desired pitch, without docking the ship; the regulating screw on the stern-shaft being suitably graduated for this purpose. Lifting-screws are now practically obsolete, and in all new single-screw steam-ships that are intended to be capable of cruising under sail alone, the propellers are fitted with feathering blades.

Mangin screw-propeller.—The general form of the Mangin screw-propeller is shown in Figs. 191 and 192. It may be regarded simply as a common two-bladed propeller in which each blade is cut in halves, with one half set immediately behind the other on the shaft, so that the width it occupies is only one-half that of an ordinary two-bladed screw. It may also be described as a four-bladed screw with the blades on opposite sides of the shaft in pairs behind each other, instead of being at right angles to each other. It will be seen that if this propeller be set in a vertical position when the ship is under sail, its projection on a transverse section would only be about one-half that of an ordinary two-bladed screw, the aggregate length of the blades being the same, so that its retarding effect on the ship would be correspondingly reduced. When the aggregate area of the four blades is equal to that of the two blades of an ordinary screw, there is little difference in the efficiency.

Some French engineers have made Mangin propellers of six blades, three in each line. When these screws were placed in a vertical position they were almost completely masked by the stern-post, and did not practically interfere with the sailing at all. It was, however, found that their

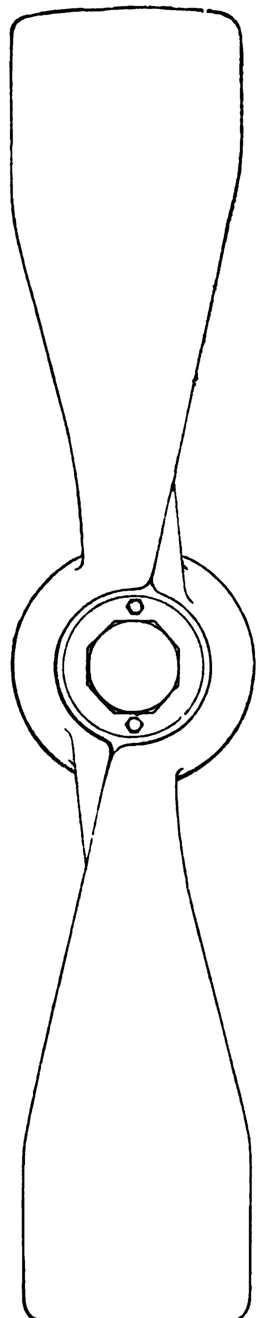


Fig. 191.

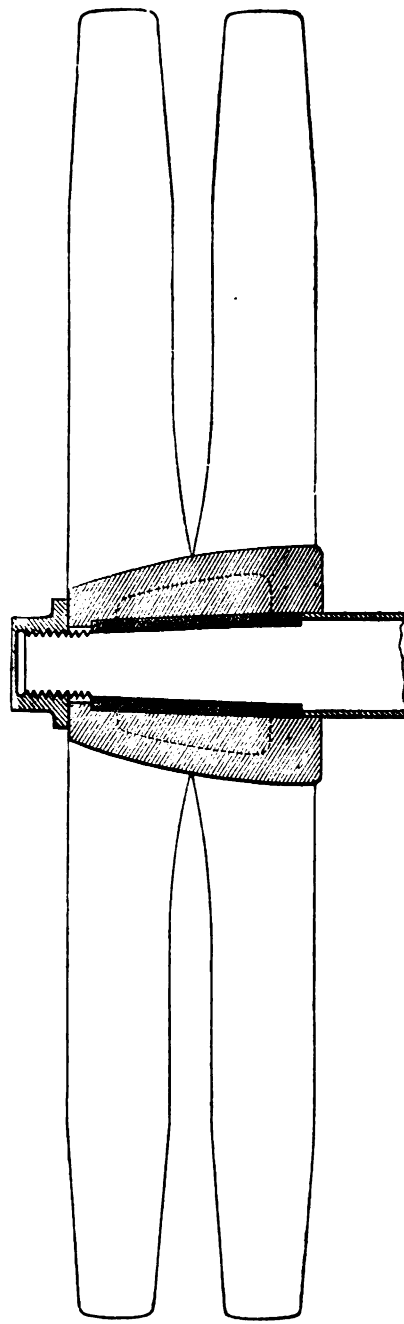


Fig. 192.

performance was decidedly inferior to that of ordinary screws, the loss of efficiency being probably due to the blades being so close to the stern- and rudder-posts and to each other. If the blades were spaced farther apart the screw would be inconveniently long, which would necessitate too great an aperture in the deadwood.

Stern-shaft and tube.—The aftermost length of the propeller-shafting, generally called the stern-shaft, which passes out through the stern of the ship to drive the screw-propeller, is covered with a gun-metal sleeve or casing. This is either cast separate, and turned, bored, and forced on when slightly warmed, or is cast around the shaft, which in this case is usually grooved, to prevent the sleeve turning. When the latter method is adopted the shaft itself should be made hot before the metal is cast around it, and the greatest care is necessary to prevent the shaft bending and to insure a sound casting. On the whole, the plan of casting the casing separately and forcing it on by a hydraulic press after it has been bored to fit the shaft is the more simple and satisfactory.

Sometimes the casing is in one piece, but more generally it is in three or more pieces, according to its length, the several parts being tightly filleted into each other at the ends, for if there should be any leakage, the shaft will decay rapidly at the joints from galvanic action. Studs should be screwed through the casing into the shaft to prevent any change in its position by the working of the engines. The parts of the casing at the ends of the shaft which work in the lignum-vitæ bearings should fit solidly on the shaft. The intermediate portion, the sole office of which is to protect the shaft from the action of the water, is generally made much lighter. In the mercantile marine, the intermediate portion of the sleeve is frequently dispensed with, the shaft only being cased with gun-metal

at the bearings, and the centre part either left bare, or lapped with wire and painted.

Stern-tube.—The stern-shaft is enclosed in a tube that extends from the inside of the ship at the after end of the screw tunnel, through the after run of the ship to the stern-post. This tube is usually known as the stern-tube and, in ships of the Royal Navy, is generally made of gun-metal. In merchant steamers it is frequently made of iron. In wood and composite ships the stern-tube is usually fitted in a wrought-iron or steel casing built into the stern of the ship, and bored out to receive it. This additional casing is also sometimes fitted in iron and steel ships.

The bearings, or rubbing parts, of the stern-tube are fitted with strips of lignum-vitæ, between which the water can pass freely to lubricate the shaft. To prevent this water passing into the ship, a stuffing-box is fitted on the bulkhead at the end of the screw-tunnel where the shaft enters the tube. When the engines are at work, the stern-gland is slackened back, to allow a little water to run through and keep the rubbing parts cool. A pipe is carried from the space between the stern-tube and the shaft, to a cock at the end of the screw-tunnel, or shaft-alley, to enable the water to be drawn off when required, to ascertain the temperature of the rubbing surfaces.

Outer thrust rings and discs.—In most modern single-screw ships, the thrust of the screw is taken directly on the stern-post by means of a ring or disc, fitted with lignum-vitæ segments. In the case of lifting screws, which are not rigidly connected to the shafting, a disc fitted with lignum-vitæ is placed in the after-bearing of the screw, between the end of the shaft and the rudder-post, to take the sternward thrust, as shown in Fig. 184. When the screw is overhung and has no after-bearing, the backward thrust

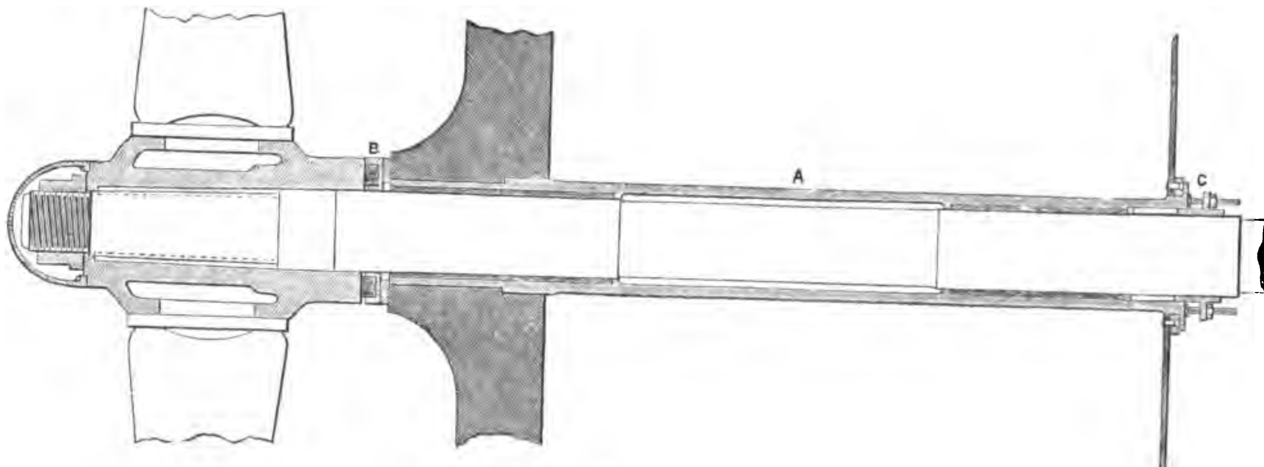


Fig. 193.

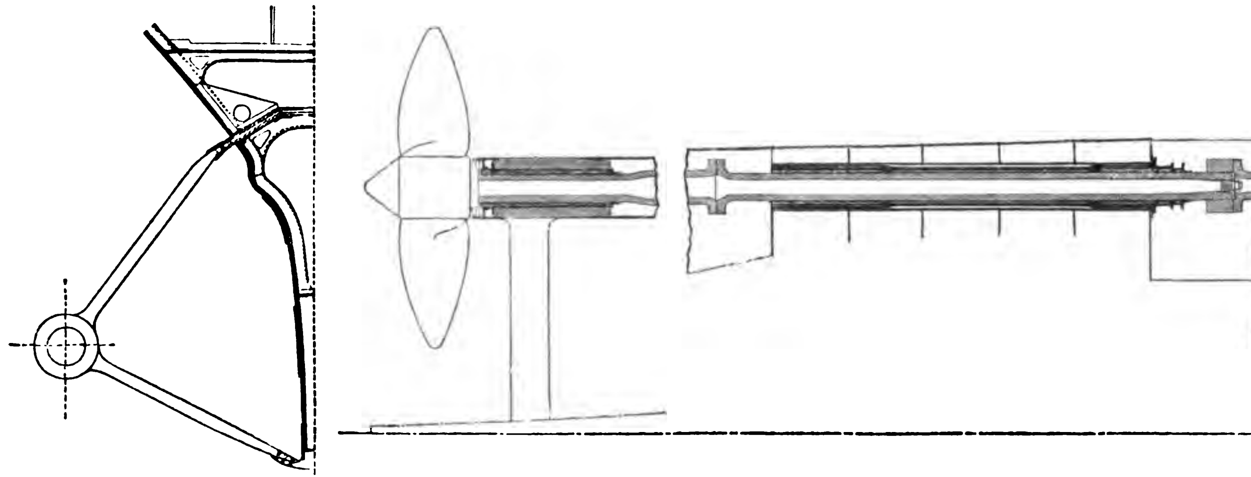


FIG. 194.

must be taken by an ordinary thrust-bearing with collars, inside the ship. Outer thrust-bearings have worked in a very satisfactory manner and are now generally adopted wherever practicable. Fig. 193 shows an arrangement of stern-shaft, tube, and fittings for an iron or steel ship with a single screw-propeller; A being the stern-tube, B the outer thrust-ring fitted with lignum-vitæ against which the forward face of the propeller-boss acts, and C the stuffing-box to prevent water entering the ship through the stern-tube.

Lignum-vitæ bearings.—The application of lignum-vitæ bearings for surfaces working in water was introduced by Messrs. John Penn and Sons, and it very greatly facilitated the progress of screw-propulsion. Originally brass bearings were used, but it was found that they wore away very rapidly. With bearings formed of lignum-vitæ strips, with water-spaces between them, it was found that a working pressure of 2,000 pounds per square inch could be safely carried, whilst in many cases metal on metal, in water, cut with not more than one-tenth of this pressure.

Twin-screws.—In twin-screw ships, the propellers are on either side of the ship under the quarters, and are usually fitted to work outwards when driving the ship ahead, and the after parts of the propeller-shafting pass outside the ship and work in bearings carried by brackets secured rigidly to the hull. In high-speed ships, in which the after run is very fine, the length of the shafting outside the ship is so great that, when of the ordinary dimensions, intermediate bearings are necessary to be fitted between the ends of the stern-tubes and the after brackets close to the propellers. In the more recent ships hollow steel shafts, of enlarged diameter between the bearings to give increased stiffness, have been fitted in order to dispense with the intermediate inaccessible bearings, and to reduce the resistance due to

the hull appendages. Fig. 194 shows the stern and propeller fittings of modern high speed twin-screw warships. In twin-screw ships the thrust of the propeller is wholly taken by the thrust bearing inside the ship, and not on discs fitted with lignum-vitæ sectors outside, as in single screw-ships.

PART VI.

GENERAL.

CHAPTER XXIV.

THÉ INDICATOR AND INDICATOR DIAGRAMS.

THE *steam-engine indicator* enables the varying pressures of the steam in the cylinder, during both the forward and backward strokes, and also the actual effective pressure at any time, to be ascertained, and the mean effective pressure on the piston during the stroke to be calculated.

The general features of the instrument are as follows :—A pencil is attached to a small piston, which works in a cylinder, open to the atmosphere at the top, and connected by means of stopcocks and pipes to either end of the engine cylinder as required. When the stopcock is open, so as to place the instrument in connection with one end of the engine cylinder, the indicator piston, carrying the pencil, is moved up and down by the varying pressures of the steam in the cylinder, the motion of the piston being opposed and regulated by the action of a spiral spring of known elastic force. The pencil traces a curve on a sheet of paper fixed on a barrel, which is caused to revolve backward and forward, in a manner coincident with the motion of the engine piston, so that a diagram is obtained from which, at any given part of the stroke of the engine, the corresponding pressures of steam in the cylinder may

be measured. The power exerted by the engines at any time is obtained by multiplying the mean effective pressure of the steam on the piston, ascertained from the diagram, by the area of the piston in square inches, and by the speed of the piston in feet per minute, and dividing the product by 33,000, the result being the *indicated horse-power* developed in the cylinder. By means of the indicator diagram, many other particulars relative to the action of the steam in the cylinders, and the adjustment and condition of the slide-valves and pistons, may be ascertained; and there can be no doubt that the great and rapid improvements that have been made in recent years in the performance and efficiency of steam-engines have been very largely assisted, and in many cases initiated, by the intelligent application of this instrument, by which the internal workings of the machinery have in a manner been exposed to view.

The following important particulars may be readily seen by inspection of the diagrams:

(1) Whether the admission of steam is early or late, the amount that the initial pressure in the cylinder is below the boiler pressure, and whether the pressure is well maintained up to the point of cut-off or not.

(2) The part of the stroke of the piston at which the admission of steam to the cylinder is cut off, and whether the cut-off is sharp or gradual.

(3) At what point and pressure the steam is admitted to the condenser.

(4) The amount of vacuum, whether obtained quickly or not, and the amount of compression at the end of the stroke.

The indicator in a crude form was invented by James Watt. Since his time its construction has been much simplified and perfected, but it is only within a comparatively recent period that its application has become

anything like general. Now, however, the regular taking of indicator diagrams is almost universal, and a vast fund of information is thereby gained, from which improvements in construction, adjustment, and management are continually suggesting themselves.

The two types of indicator now most generally used are known as McNaught's and Richards' indicators. In the Royal Navy the former is issued for general use on ordinary service, but for the records of steam trials, and other special services, the Richards indicator is employed. We will therefore describe both instruments, and point out the features in which they differ from each other.

McNaught's indicator.—The general form and arrangement of McNaught's indicator is shown in Fig. 195. A B is a cylindrical case. Its lower end A contains a small cylinder, of sectional area equal to one-quarter of a square inch, in which a piston works steam-tight, but with as little friction as possible. To the lower end of this cylinder a straightway cock C is fitted, having a small hole bored in its side to connect the under side of the indicator piston with the atmosphere when the cock is shut. The end of this cock is screwed to enable it to be attached to the nozzle of a right-angled 3-way cock, called the *indicator cock*, which is connected by pipes to the two ends of the engine cylinder, so that the indicator may be placed in communication with either side of the piston as desired.

This is the usual fitting, and is for the convenience of enabling the two diagrams showing the pressures of the steam on both sides of the piston to be taken on one card, but, if preferred, the indicator might be connected directly to each end of the cylinder, so as to get separate diagrams, and this course is sometimes desirable in very quick-moving engines, especially in cases where a fair lead of pipes cannot be readily obtained to enable a combined diagram to be taken.

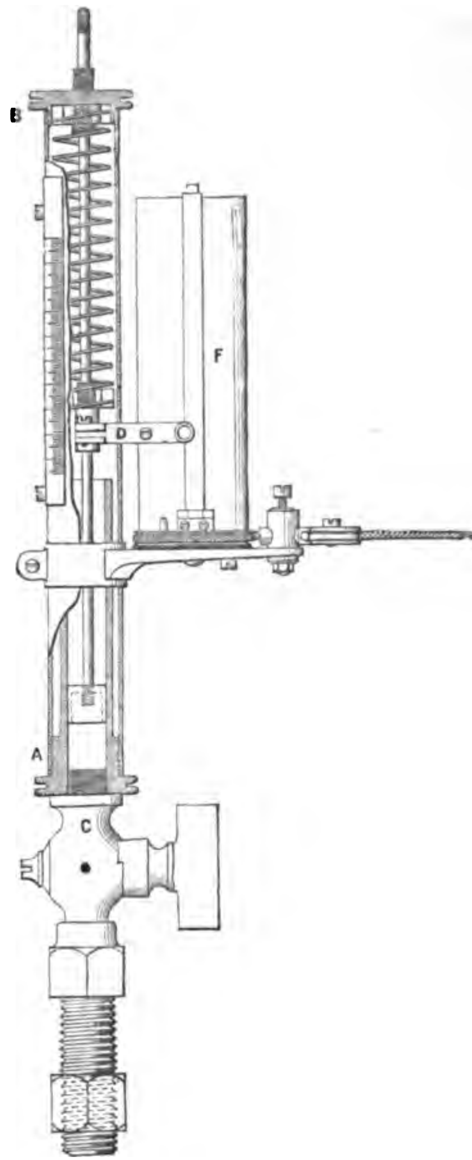


FIG. 195.

The upper side of the piston is open to the atmosphere; and a spiral spring of known tension, attached to the piston-rod and also to the top of the casing, resists the motion of the piston, when acted on by the steam pressure underneath, or by the atmospheric pressure above when the pressure of steam in the cylinder falls below that of the atmosphere. The tensions of the springs generally employed in the Service are 10, 15, 30, and 50 pounds to the inch. If, for example, a 30-pound spring be fixed in the instrument, a steam pressure of 30 pounds per square inch will compress the spring through a distance of one inch, so that the actual pressures of steam are measured, according to the proper scale, by the height to which the piston is moved, every vertical inch on the diagram representing a steam pressure of 30 pounds per square inch on the engine piston. The tension of the spring selected for use in any particular case will depend on the working pressure of the steam, the higher pressures requiring the stiffer springs. In ordinary practice,

50-pound springs are used for pressures above 70 pounds per square inch.

30-pound springs are used for pressures between 35 pounds and 70 pounds per square inch;

15-pound springs are used for pressures between 22 pounds and 35 pounds per square inch;

10-pound springs are used for pressures below 22 pounds per square inch.

A short arm D, projecting from the indicator piston-rod, carries the pencil with which the diagram is traced. This arm is jointed so that the pencil may be pressed on, or taken off from, the paper as required. The pencil is generally a small brass wire, the paper being specially prepared to enable the metal to mark it. F is a brass drum on which the diagram paper is wrapped and held by clips. This paper cylinder is caused to revolve around a vertical axis by means

of a cord attached to any reciprocating part of the engine that has the same motion as the engine piston: the extent of motion being suitably reduced by means of levers, etc., to cause the paper cylinder to make about three-fourths of a revolution for each stroke of the engine. The tension is kept on the string, and the paper cylinder brought back to its original position as the engine piston returns, by the action of a spiral spring inside the cylinder F.

In order to take a diagram, after the instrument has been fixed to the indicator cock, the string is first con-

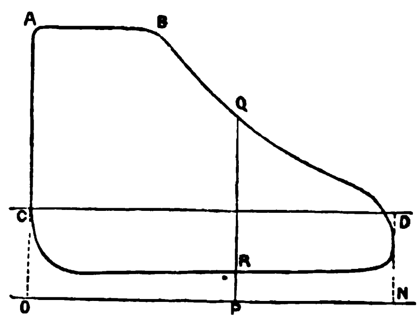


FIG. 196.

nected to the indicator lever, and its length adjusted by means of a running loop, to give the proper movement to the paper cylinder. When the cock at the bottom of the indicator is shut, the atmosphere has free access to both sides of the indicator piston, and keeps it in equilibrium. In this position the pencil, on being pressed against the revolving paper, will trace a straight line, which will represent the atmospheric pressure, and is called the atmospheric line of the diagram. This is marked C D in Fig. 196.

If the cock C on the instrument be opened, and the indicator cock set to make connection with one end of the engine cylinder, the pencil will move up and down according to the variation of pressure in the cylinder

during the stroke, and trace out a curve similar to Fig. 196 during each complete stroke.

This diagram shows the action of the steam on one side of the piston only, during a complete revolution. To ascertain the action on the other side of the piston, the indicator cock must be placed in communication with the opposite end of the cylinder, when a similar diagram, but reversed, will be obtained. If on the diagram a horizontal line be drawn at a distance below C D, equal, on the given scale, to the atmospheric pressure, say 14·7 pounds per square inch, this line O N will be the zero line, or line of no pressure, and all ordinates measured from this base line will represent *absolute pressures* of steam per square inch. For example, if we take the total length O N of the diagram to represent the length of stroke of the piston of the engine, then at the part of the stroke represented by the point P, the forward absolute pressure on the piston will be P Q, and the back pressure P R, the effective forward pressure being equal to P Q $-$ P R $=$ Q R. P R is actually the back pressure for the return stroke, but it has been assumed for illustration that the back pressure is the same for both sides of the piston.

In the McNaught indicator the extent of the motion of the pencil is the same as that of the indicator piston, so that with high-pressure steam the springs have to be compressed to a considerable extent, almost instantaneously, on the admission of steam to the cylinder. In quick-moving engines, this often causes much oscillation and vibration in the motion of the pencil, and a series of undulations are formed on the diagram, which detract from its value as an indication of the action of the steam in the cylinder. A check screw and guard are usually fitted on these instruments to prevent this vibration as much as possible.

Richards' indicator.—The Richards' indicator was

introduced to remedy this defect, and its application has been very successful, more particularly in cases of engines working with high pressures and rates of expansion, from which it is difficult to obtain satisfactory diagrams with the common indicator. In principle it is the same as

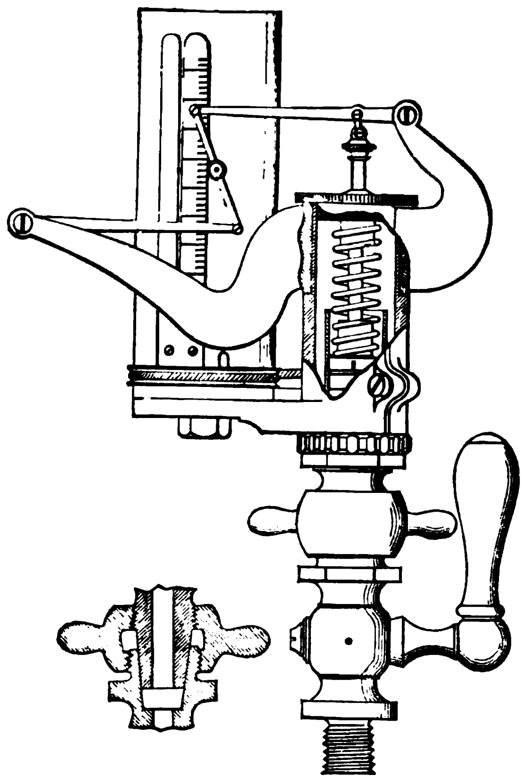


Fig. 197.

McNaught's, and the method of its application is the same; the only difference being in the details of its construction, by which excessive oscillation of the pencil is prevented.

The general form and arrangement of the Richards'

indicator is shown in Fig. 197. In these instruments the area of the piston is half a square inch, and the pencil, instead of being attached directly to the piston-rod, is worked by a lever of the third order, its extent of motion being four times as great as that of the piston. The maximum length of stroke of the piston is $\frac{25}{32}$ inch, which gives an extreme travel of pencil of $3\frac{1}{8}$ inches. The stronger springs required for these instruments and the less extent to which they have to be compressed by the steam render them less liable to oscillation; and as a matter of fact, diagrams have been taken by them from locomotive and other high-pressure quick-moving engines without showing any appreciable vibration. The parallelism of the pencil is maintained by an ingenious arrangement of light steel rods, carried by a brass bracket fitted on the top of the cylinder, round which it can be rotated by hand, as required, to bring the pencil on or off the metallic paper wound on the barrel. The details of the arrangement are clearly shown in Fig. 197.

Having thus described the instruments, and the method of their application, we will pass on to consider the indicator diagrams themselves and explain the manner in which the performance of the engine can be deduced from them. Actual indicator diagrams present so many differences in form, resulting from the varying action of the steam in the cylinder under different circumstances, that it is impossible to exactly represent the action mathematically; especially as we are unable to accurately separate the irregularities in the diagram due to real fluctuations in the pressure of the steam, from those due to mechanical causes, such as friction of the indicator piston, and want of adjustment of its parts, etc.

The question will be simplified and a clearer knowledge of the nature of the indicator diagram gained if, in the first place, we consider the theoretical diagram, which

was explained at page 159, Chapter IX., and afterwards point out the deviations of actual diagrams from this theoretical diagram, resulting from various causes. This ideal diagram is similar in form to the actual diagram, but much more simple in construction. The length of the indicator diagram is generally, in practice, taken to represent the stroke of the piston, so that the ordinate drawn at any point of the diagram represents the pressure at the corresponding portion of the stroke.

The theoretical indicator diagram is shown in Fig. 198.

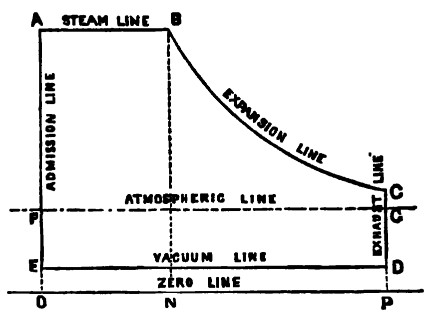


Fig. 198.

O A is called the admission line of the diagram
A B „ „ steam line „ „
B C „ „ expansion line „ „
C D „ „ exhaust line „ „
D E „ „ vacuum or back-pressure line „ „
F G „ „ atmospheric line „ „
O P „ „ zero line, or line of no pressure.

The assumptions made in forming the theoretical diagram are:

1. That the full pressure of steam acts suddenly on the piston at the beginning of the stroke.

2. That this initial pressure remains constant up to the point of cut-off.

3. That the expansion is continued up to the end of the stroke.

4. That the communication with the condenser is suddenly opened at the end of the forward stroke, the pressure falling suddenly to that in the condenser.

5. That the cylinder is open to the condenser during the whole period of the return stroke of the piston, the back pressure remaining constant.

We will now point out the deviations from the theoretical diagram that must necessarily exist in an indicator diagram

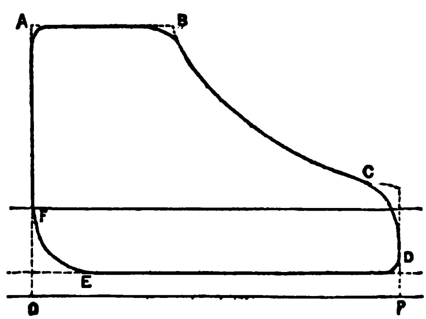


FIG. 199.

taken from an actual engine in good order, the various parts being properly fitted and adjusted. The cylinder ports and passages are supposed to be sufficiently large to prevent *wire-drawing* of the steam during admission, so that the steam line will still be approximately horizontal.

Wire-drawing the steam.—Wire-drawing is the technical name for the reduction of pressure which steam undergoes by its passage through contracted areas, and by which its efficiency is always reduced.

With ordinary slide or expansion valves the cut-off is not absolutely sudden, as is assumed in the theoretical

diagram, but gradual; so that instead of having a point at B, the actual diagram would be somewhat rounded, as shown in Fig. 199. With expansion valves fitted specially for early points of cut-off, the rounding of the corner is less than with the ordinary slide-valve, but it must always exist to a greater or less extent.

After the cut-off, the steam expands in the cylinder, as assumed in the ideal diagram; but in actual engines the communication with the condenser is opened before the end of the stroke, say at nine-tenths to eleven-twelfths of the stroke, and at this point the pressure begins to fall rapidly till the end of the stroke, making the obliquely-curved line C D. This opening before the end of the stroke is necessary to insure the vacuum being nearly complete when the piston commences its return stroke; and since at the time of release the piston is near the end of its stroke, and consequently moving very slowly, the forward pressure is very little affected by the early release.

During the greater part of the return stroke, the cylinder is open to the condenser, and the back pressure is practically constant; but when the piston arrives at some point E, before the end of the stroke, the communication with the condenser is closed, and the residue of steam in the cylinder is compressed behind the piston until just before the end of the stroke, its pressure rising to F, when fresh steam enters and a new stroke commences. This action is called *compression* or *cushioning*, and has the effect of preventing shock to the mechanism, by the sudden admission of steam of the full pressure; it also tends to reduce the loss from clearance (see p. 501). The corner E F of the diagram is called the compression or cushioning corner.

We see, therefore, that the form of the indicator diagram taken from an actual engine in good order and properly adjusted is not very different from that of the ideal

diagram, the principal deviations being in the rounding of the corners of the diagram and in the existence of compression or cushioning at the end of the return stroke. The main points, therefore, that distinguish a good diagram are:

1. Steam and vacuum lines horizontal: more especially the latter, as will be pointed out further on when considering the faults of diagrams.

2. The expansion curve clear and regular, free from vibration, and approximating to a rectangular hyperbola.

3. The cushioning corner not too great, but yet sufficient to prevent jars or shocks to the machinery.

In considering the faults of diagrams, it must be clearly understood that the only facts absolutely given by the indicator diagram are, the actual pressures of the steam on the piston, and the manner in which the pressure varies during the stroke. In order to determine the causes of these variations and draw reliable conclusions from the diagrams, a correct knowledge of the action of the steam in the engine is necessary. The study of the forms of diagrams, taken under different circumstances, will be found to be most interesting and instructive; and if carefully pursued, many abnormal features in the diagrams will be found to possess peculiar value, and many improvements in the engine may be confidently suggested. It is, however, essential that the deductions drawn should be correct, and careful study is required to enable the information contained in the diagram to be properly understood and turned to practical advantage. We shall only be able to indicate a few of the more important and general causes affecting the form of the diagram. The interpretation of the minor and abnormal sources of variation must be left to practical study of actual diagrams. It is, however, believed that the short sketch given will enable the majority of indicator diagrams to be easily and correctly interpreted.

Effect of insufficient opening for the admission of steam.—In this case, suppose O A to be the initial absolute pressure in the cylinder. In consequence of the restricted area of the opening, the steam will not be able to follow the piston fast enough to keep up its full initial pressure, and will consequently be wire-drawn: the steam line, instead of being horizontal, will slope downwards as A B, Fig. 200. The efficiency will consequently be reduced, as the expansion will be due to the pressure N B only, instead of N E as shown by the dotted line; so that the loss of work by wire-drawing the steam on admission is

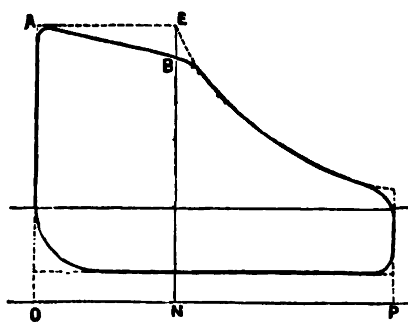


FIG. 200.

nearly equivalent to a reduction of the initial pressure from O A to N B.

In consequence of practical difficulties in arranging the slide-valves and cylinder ports of large quick-moving engines, this defect may often be noticed in their full-power diagrams, especially when the steam is carried for a considerable portion of the stroke. With an early cut-off, it is not so likely to occur, as the motion of the piston is comparatively slow for the earlier portion of the stroke. When this wire-drawing only exists on the steam side, it is not of so much importance, and in many cases is practically unavoidable, though, as we have seen, it involves a

loss of efficiency; but in order to prevent a similar action during the exhaust, which would be much more serious, it is usual to make the exhaust ports and passages at least 50 per cent. greater than the steam openings. In Fig. 200, the only defect is the wire-drawing of the steam during admission.

Steam and exhaust openings both contracted.—This is the most serious defect that can be shown by a diagram, as it is one that is generally due, not merely to incorrect adjustment of the parts, but to faulty construction, and can only be remedied by enlarging the cylinder ports and

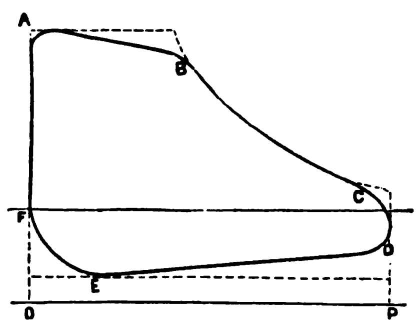


FIG. 201.

passages, which, in most cases, would involve the substitution of new cylinders.

Fig. 201 is an illustration of the kind of diagram that might be expected in this case; somewhat exaggerated for the purpose of illustration. The action on the steam side would be similar to that previously described and shown in Fig. 200. But, in addition, in consequence of the restricted area of the exhaust passages, the steam will not be able to escape freely from the cylinder to the condenser, so that the back pressure will be increased, and the vacuum not fully attained until the piston has traversed some portion of its return stroke, the vacuum line sloping downward from D to E instead of being

horizontal. It is evident that wire-drawing during the exhaust is considerably more injurious than during admission, as it affects nearly the whole length of the diagram during the return stroke, increasing the average back pressure, and thereby reducing the power of the engine. It is not often that defects of this nature exist to so great an extent as that shown in Fig. 201, which, however, illustrates the character of the indications by which this defect would be shown on the diagram.

Cylinder ports and passages.—In the design of engines, the cylinder ports and passages should be made sufficiently large to prevent the fall of pressure during admission to as great an extent as possible. The proper proportion that the area of the ports and passages should bear to that of the cylinder will depend upon the pressure of the steam and the speed of the piston. In general practice the area of the steam ports at each end of a cylinder varies from $\frac{1}{6}$th to $\frac{1}{12}$th the area of the cylinder: the larger sizes being used for the faster running engines. The following approximate rule may be used for calculating the port area in any particular case:—

$$\frac{\text{area of port}}{\text{area of piston}} = \frac{\text{speed of piston in feet per minute}}{7{,}000 \text{ to } 8{,}000}.$$

It is most important to arrange for the free and unrestricted exhaust of the steam from the cylinder at the end of the stroke, for any defect in this action causes a much more serious reduction of power of the engines than would arise from deficiency in the area of opening for admission alone.

Eccentric set with insufficient advance so that the communication to the condenser is not opened until the piston reaches the end of the forward stroke.—In this case, when the steam begins to be released from the cylinder, the piston has already begun its return stroke, and consequently the pencil of the

indicator has commenced to trace the back-pressure line. The expansion curve will therefore be carried out to the end of the diagram, and the exhaust line will curve from C to D, as shown in Fig. 202, the piston traversing a portion of its return stroke before the full vacuum is attained. It must also be borne in mind that the points of release and compression will vary with each other, so that in this case there would be little or no 'cushioning,' and the left-hand corner of the diagram will be nearly square. The admission will also be late, and the line will slope inwards as F·A.

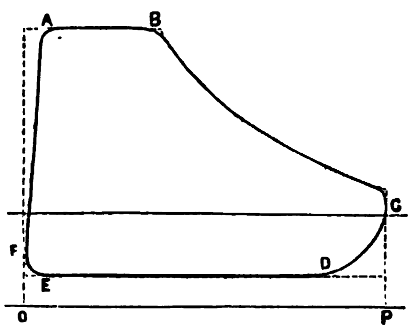


FIG. 202.

Eccentric so set on the shaft that the slide-valve opens the communication with the condenser too early.—The defects in this diagram will be of the opposite character to that shown in Fig. 202, all the operations being too early. Since the release is early the compression will be correspondingly early, forming a large cushioning corner, and the form of the diagram will be as shown in Fig. 203. The release being early, the full vacuum will be attained at the commencement of the return stroke, the exhaust line sloping as shown by C D on the diagram; there will also be a large cushioning corner, as shown by E F. This form of diagram is given when the steam is worked expansively by the use of the link motion. The

effect of 'shortening the link,' as it is technically called, is to shorten the stroke and increase the lead of the slide-valve, by which the actions of admission, cut-off, release, and compression are all made earlier.

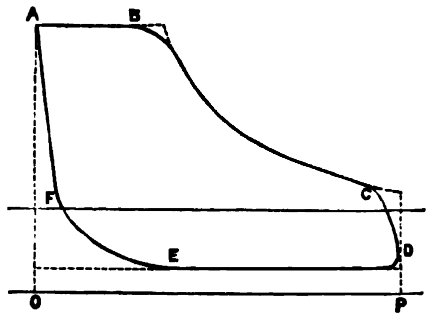


FIG. 203.

Leaky slide or expansion valves.—If the valves are leaky, steam will continue to enter the cylinder after the admission is nominally cut off. In consequence of this the expansion curve, instead of falling regu-

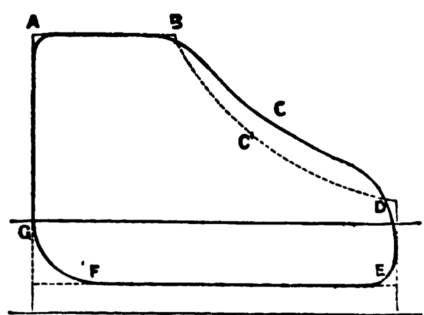


FIG. 204.

larly, and approximating to the hyperbola, as in the dotted line B C′ D, Fig. 204, which would be the case if the valves were tight, would rise above this curve, as shown by the full line B C D, the pressure being increased

by the leakage during expansion. This clearly, in any case, would involve a waste of steam and loss of efficiency, and should be carefully guarded against, especially in slow-moving engines, in which the percentage of loss would be the greater.

Leaky Pistons.—This defect would be shown by the vacuum line of the diagram. If the piston leak, steam will pass from the steam to the exhaust side of the piston during the stroke, which would increase the back pressure and tend to make the vacuum line irregular, as shown in Fig. 205. This is a very serious defect, because the steam which thus passes the piston goes direct to the condenser without doing any useful work, and is not only wasted, but, in addition, reduces the efficiency

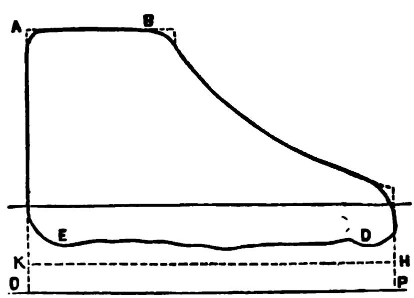


FIG. 205.

of the engine by increasing the back pressure opposing the motion of the piston. In a compound engine any steam leaking past the piston of the high-pressure cylinder would have a chance of doing useful work in the low-pressure cylinder, so that the loss would not be so great.

So far, for simplicity, we have considered the defects in diagrams taken from one side only of the piston. We will now explain how the effects of wrong adjustment are shown on the combined diagram taken from the two sides of the piston.

Suppose the slide-valve incorrectly set on the rod, so that the leads at the opposite ends of the cylinder are not equal to each other: the eccentric being set at the proper angle of advance.—In this case one end of the valve will have insufficient lap on the steam side and too much on the exhaust, and at the other end of the valve the errors will be of an opposite character. The valve will therefore at one end admit steam too early, continue the admission too long, and cut off too late, while at the opposite end the operations will be reversed. On the exhaust side, the valve, at the end with early and lengthened admission, will commence to exhaust late and the period of the exhaust will be shortened; at the other end the exhaust will begin

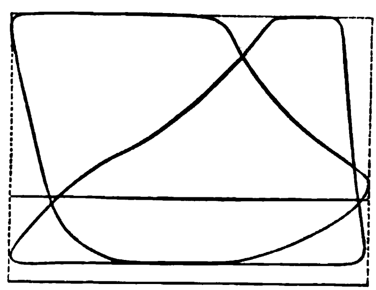


FIG. 206.

early and continue for an increased portion of the stroke. The diagram will therefore be of the character shown in Fig. 206.

Suppose the slide-valve correctly set, that is, with equal lead at each end, but the eccentric secured in a wrong position on the shaft.—If the angle of advance be too great, all the operations on *both* sides of the piston will be too early, the diagram for each stroke being similar to Fig. 203, so that the combined diagram will be as shown in Fig. 207. The admission lines both slope outwards, and both compression corners are large.

If the eccentric be fixed with too small an angle of advance, all the actions of the slide-valve will be late, and the diagram from each side of the piston will be similar to Fig. 202, the combined diagram being as shown in Fig. 208. The faults in this case are clearly of the opposite

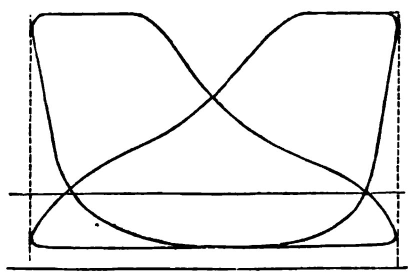


FIG. 207.

character to those shown in Fig. 207, the admission lines sloping inwards and both compression corners being small.

In both these examples, however, the faults in the figures, from the opposite sides of the piston, are similar

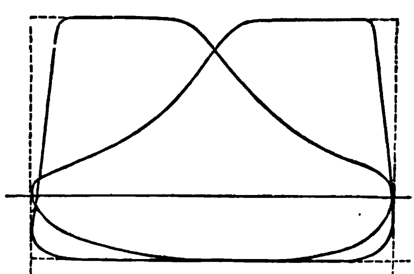


FIG. 208.

in character, and not opposite as in the case of the valve being wrongly placed on the rod. These defects may be remedied by correcting the position of the eccentric on the shaft, and setting it with the proper angle of advance, to give the slide-valve the required lead, etc.

We may therefore state generally:

1. That when the eccentrics are fixed in their proper positions on the shaft, but the slide-valve incorrectly set on the rod, the faults in the indicator diagrams from *opposite* sides of the piston will be of *opposite* natures.

2. That when the eccentrics are wrongly keyed on the shaft, so as to give either too great or too small an angle of advance, the faults in the indicator diagrams from *opposite* sides of the piston will be *similar* in character.

All the defects in diagrams hitherto discussed affect generally, not only the form of the diagram, but also the power and efficiency of the engine. We will now mention a few causes which affect the form of the diagram but not the power of the engine. These may be briefly summarised as follows:

1. Undulation or vibration of the pencil.
2. Friction of the indicator.
3. Position of the indicator.
4. Length of the string.

The first is principally due to the weakness of the springs, especially when of considerable length. The steam, on admission, exerts its full pressure on the indicator piston, and the inertia of the piston causes a series of oscillations in the pencil, which traces an undulating figure on the card. An example of this is given in Fig. 209, and this action may in some cases be so violent as to render the diagram of little value as an indication of the action of the steam in the cylinder. In case of undulations, it will be more correct to calculate the horse-power from a dotted line drawn midway between the crests and hollows, as shown in Fig. 209, than from the actual diagram itself.

Friction of the indicator.—When this occurs it opposes the motion of the indicator piston, and therefore tends

to make the indicated forward pressure less and the indicated back pressure greater than is correct, and so to make the indicated work appear less than is really exerted. In practice, if the instruments are kept in order, this will not exist, because the indicator pistons are always made with a very slight degree of leakage, so as to make them as nearly as possible frictionless. Sometimes, however, dirty matter is carried into the indicator cylinders with the steam, which would increase the friction, and it is necessary that the instruments should be fre-

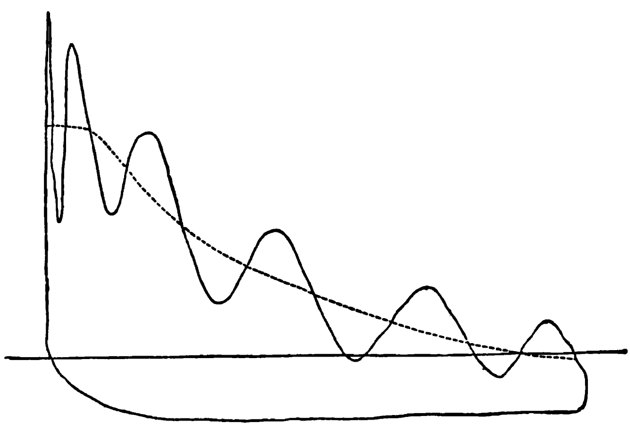


FIG. 209.

quently examined and cleaned, to insure correct results being obtained. In cases of high expansion, if the indicator piston be too tight, the defect is sometimes shown by a series of steps on the diagram, the piston, instead of following the steam freely, descending in jumps in consequence of the friction.

Position of the indicator.—If the position of the indicator is such that a rapid current of steam passes across the nozzle, the steam pressure shown on the diagram will be thereby reduced. Sudden bends, great length and smallness of diameter in the indicator pipes,

also tend to reduce the indicated pressure given by the diagrams. The bore of the pipes for a Richards' indicator should not be less than $\frac{1}{2}$-inch for the straight parts and $\frac{5}{8}$-inch in the bends.

Length of string.—The length of the string should be carefully adjusted before taking the diagram, for if the string be either too long or too short, the paper cylinder will come to rest before the piston reaches the end of its stroke. The pencil will consequently trace a vertical line when it should be inclined, which will cause the corners to be square and incorrect, and the effect will be the same as if a vertical line were drawn, cutting off a portion of the proper diagram.

On all indicator diagrams there should be marked the scale of the diagram, the amount of steam pressure and vacuum shown by the gauges, and the number of revolutions per minute the engine was making at the time the diagram was taken.

Calculation of indicator diagrams.—By means of the indicator diagram we are enabled to estimate the mean effective pressure on the piston during each stroke; and consequently, the area of the cylinder, length of stroke, and number of revolutions per minute, being known, the indicated horse-power of the engine can be readily calculated.

If p_m = mean effective pressure on the piston in pounds per square inch.

and A = area of piston in square inches:

then $p_m \times A$ = total pressure on the piston in pounds;

and if S = length of stroke in feet,

N = No. of revolutions of the engines per minute:

then $2 S \times N$ = speed of the piston in feet per minute.

Therefore, $p_m \times A \times 2\,S \times N =$ the number of foot-pounds of work developed per minute.

Consequently, since one indicated horse-power is equal to 33,000 foot-pounds, the indicated horse-power developed in the cylinder in question

$$= \frac{p_m \times A \times 2\,S \times N}{33,000}.$$

If there be more than one cylinder, the powers developed in the several cylinders must be added together to obtain the total indicated horse-power.

The following is the method generally adopted practically for ascertaining the mean effective pressure from an indicator diagram.

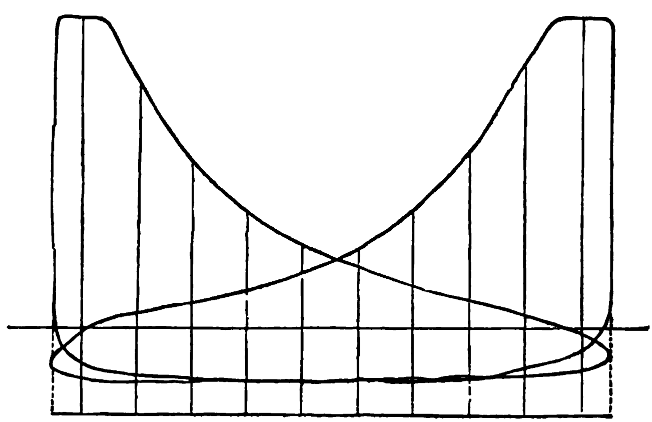


FIG. 210.

The total length of the diagram is divided into ten equal parts, and vertical ordinates are drawn at the middle points of the spaces thus formed. The first and last ordinates will then each be $\frac{1}{20}$th of the length of the diagram from the end, and the common distance between the several ordinates will be equal to $\frac{1}{10}$th of the length of the diagram. This method of division is shown in Fig. 210.

On the diagram taken from *each* side of the piston, the lengths of the several ordinates intercepted between the forward and back pressure lines are measured on the required scale, added together, and divided by 10. The result gives the mean effective pressure on one side of the piston. The same process is carried out on the other diagram, to find the mean effective pressure on its side of the piston, and the mean of the two means thus obtained gives the *mean effective pressure* for the complete double stroke of the engine, which is used in calculating the indicated horse-power.

To facilitate calculation, the lengths of the ordinates are usually measured successively on a strip of paper, the second ordinate commencing at the end of the first, and so on. The total length thus obtained will clearly be the sum of the lengths of the 10 ordinates, and this may be measured on the scale at one operation, and divided by 10 to give the mean pressure. This process of adding the lengths geometrically instead of by figures, besides saving time and labour, is more correct, as it avoids multiplying the small errors that may exist in the measurement by scale of each of the ordinates separately, and adding them arithmetically. The scale on which the strips are measured is often made ten times as great as the scale of the diagram, so that the mean pressure may be read off, without requiring the division by 10.

In the right-hand side of the equation

$$\text{I.H.P.} = \frac{p_m \times A \times 2\,S \times N}{33{,}000}.$$

p_m and N are the only variables, for the same cylinder. It is therefore usual in practice to combine the constant quantities to further facilitate calculation. The *cylinder constant* for any engine is clearly

$$= \frac{A \times 2\,S}{33{,}000} = C, \text{ suppose.}$$

This constant, multiplied by the mean pressure calculated from the diagram, and by the revolutions of the engine per minute, will give the indicated horse-power of the cylinder,

$$\text{or, I.H.P.} = C \times p_m \times N.$$

Clearance.—The form of the expansion curve is very much affected by the clearance at the ends of the cylinders. The term clearance must be understood to include not only the clearance proper, or the space between the piston and the end of the cylinder when the piston is at the end of the stroke, but also, in addition, the volume of the passages between the slide-valve and the cylinder; for it is evident that these spaces must be filled with steam at the instant of cut-off; and the expansion, instead of being due simply to the volume of steam in the cylinder itself at the point of cut-off, is due to that in the cylinder plus that in the cylinder ports and passages up to the face of the slide-valve. For convenience, this clearance is represented as a fraction of the length of the stroke, and it varies considerably in different engines.

If A = area of the piston,
S = length of the stroke.

Then, $\dfrac{\text{volume of clearance space}}{A \times S}$

$= c =$ fraction of the length of stroke represented by the clearance.

If S be in inches, then cS will represent the number of inches of length of stroke equivalent to the clearance.

On an indicator diagram this may be shown in the following manner.

Let A B C D E (Fig. 211) represent an indicator diagram, and O P the length of the stroke. Take O O′ to represent the fraction of the stroke equivalent to the clearance, and draw O′ A′. Then the expansion curve will be due to the

volume of steam represented by A′ B, and not by A B, the steam line of the diagram.

An example showing very clearly the effect of the clearance on the expansion curve, in an extreme case, is

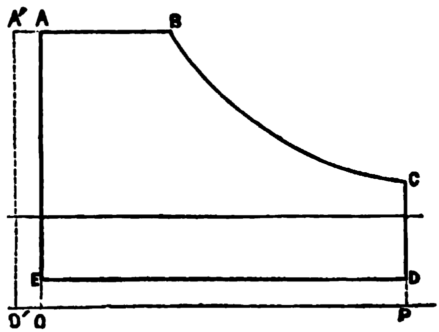


FIG. 211.

shown in Fig. 212, which is a copy of a diagram taken from one of the high-pressure cylinders of a four-cylinder

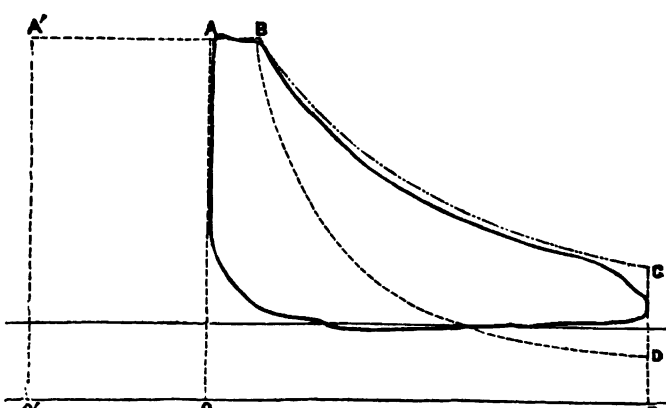


FIG. 212.

compound engine of the type shown in Fig. 159. The slide-valves of both the high- and low-pressure cylinders

are worked by the same rod, so that the clearance spaces in the high-pressure cylinders are unusually large. In the present instance the clearance represented 12 inches of length of the cylinder, or two-fifths of the entire stroke. By setting off O O′ to represent the clearance space and drawing O A′ vertical, we see that the expansion is due to the volume of steam represented by A′ B, the dotted curve B C being the theoretical expansion line for that volume. If there were no clearance, so that the expansion would be due to the volume represented by the steam line A B of the indicator diagram only, the theoretical expansion curve would be shown by the dotted line B D and the final pressure would be represented by P D instead of P C. This diagram shows the importance of taking the total amount of the clearance spaces of the cylinder into account in considering the indicator diagrams of any engine.

The effect of the clearance on the expansion may be estimated from Fig. 211.

$$\frac{\text{D E}}{\text{A B}} = r' = \text{apparent rate of expansion.}$$

$$\text{Therefore } \frac{\text{A B}}{\text{D E}} = \frac{1}{r'}.$$

$$\text{The actual ratio of expansion, } r = \frac{\text{O}' \text{ P}}{\text{A}' \text{ B}}.$$

$$\text{Therefore } \frac{1}{r} = \frac{\text{A}' \text{ B}}{\text{O}' \text{ P}} = \frac{\text{A B} + \text{O}' \text{ O}}{\text{D E} + \text{O}' \text{ O}}$$

$$= \frac{\frac{1}{r'} + c}{1 + c}$$

$$r = \frac{\text{O}' \text{ P}}{\text{A}' \text{ B}} = \frac{1 + c}{\frac{1}{r'} + c} = \frac{r' (1 + c)}{1 + c r'}.$$

For example, suppose the apparent ratio of expansion of steam in the cylinder, as shown by the cut-off gear, to

be eight times, and the total clearance spaces to be one-eighth of the total capacity of the cylinder, which is not an excessive value:

Then the actual rate of expansion,

$$r = \frac{r'(1 + c)}{1 + cr'}$$

$$= \frac{8(1 + \frac{1}{8})}{1 + 8 \times \frac{1}{8}} = \frac{9}{2} = 4\tfrac{1}{2};$$

so that the actual rate of expansion of steam will only be $4\frac{1}{2}$ times, instead of 8 times as indicated by the position of the piston at the point of cut-off.

The effect of clearance is to diminish the efficiency of the expansion and cause waste of steam; for not only the cylinder itself, but all the clearance spaces, must be filled with fresh steam each stroke. In cases of high expansion in a single cylinder, unless special care be taken to reduce the clearance as much as possible, the loss from this cause may become very considerable. The deposition of moisture in the clearance spaces may also vitiate the vacuum in the cylinders, by its boiling off during the period of exhaust and rendering the condensation of the steam less perfect.

The waste of steam resulting from clearance is reduced by the compression at the end of the return stroke, and if the compression be so great as to raise the pressure in the clearance spaces, just before the point of admission, to the initial pressure of the steam, loss from clearance would obviously be prevented; but it is evident that the mean pressure of the steam during the stroke would be reduced by the excessive amount of cushioning required, and it is not generally convenient to work with so great an amount of compression as this in actual practice. In cases, however, in which engines are worked at very high rates of expansion in a single cylinder, it is always

advantageous to use a high degree of compression, in order to reduce loss from clearance, as well as to prevent shock to the machinery on the change of the stroke by the sudden admission of steam of high pressure to the cylinder.

Clearance with gridiron expansion-valves.—In engines fitted with gridiron expansion-valves working on the tops of the slide-casings, the effect of clearance when using high rates of expansion is sometimes marked most clearly on the indicator diagrams.

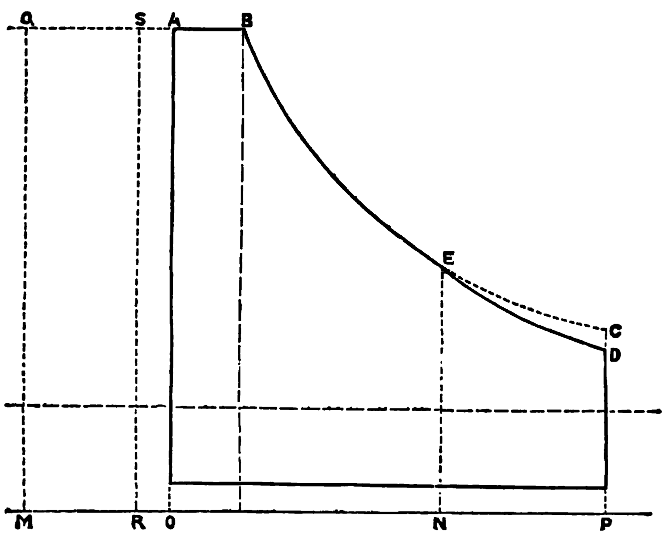


FIG. 213.

In Fig. 213, let B represent the point of cut-off by the expansion-valve. The steam then occupies all the volume of the slide-casing, cylinder ports and passages, &c. up to the piston. If O P represent the length of the stroke, and O M the fraction of the length equivalent to the clearance up to the expansion-valve (which will of course include the volume of the slide-valve casing), the expansion of the

steam at first will be due to the volume Q B. When the slide-valve closes the steam-port, say at some point N of the stroke, the clearance for the remainder of the stroke will only be the volume of the passages, &c. up to the face of the slide-valve. Let this be represented by O R; then the remainder of the expansion will clearly be due to the volume R N instead of M N, which will cause a change in the form of the curve, as shown in the diagram, the final pressure being represented by P D instead of P C. The dotted line E C is the continuation of the expansion curve for the earlier part of the stroke. On account of the excessive clearance spaces, gridiron valves so situated are not efficient for carrying out high rates of expansion in a single cylinder.

Water used per indicated horse-power.—From the indicator diagram the quantity of steam, and consequently of water, used per indicated horse-power per hour may be ascertained approximately. The values calculated from the diagrams are not absolutely correct, and are always less than the actual quantities of steam used, because they do not include the waste of steam due to liquefaction in the cylinders, radiation, and other causes. Relatively, however, as showing the differences in the performance of different engines, or of the same engine working under different circumstances, this application of the diagram is of considerable value, and much useful information may be thus gained.

In order to calculate the quantity of steam used in the engine, it is necessary in the first instance to draw on the indicator diagram the zero line, or line of no pressure. The distance of this below the atmospheric line will depend on the height of the weather barometer when the diagrams were taken. For practical purposes, however, it will be sufficiently accurate to draw this line at a distance below the atmospheric line equal, on the scale of

the diagram, to 14·7 pounds. The volume of the clearance at each end of the stroke should be ascertained and expressed as a fraction of the length of the stroke.

Take a point on the diagram, *just before* the communication is opened to the condenser. Suppose this to be when the piston has travelled d inches, and let c be the number of inches of the stroke equivalent to the clearance.

Let p be the absolute pressure of the steam at the given point of the stroke, which should be as near as possible to the end, but *must* be before release commences;

and A = area of the piston in square inches.

Then for each stroke we shall have

$A \times (d + c)$ cubic inches of steam used, at absolute pressure of p pounds per square inch.

If N = No. of revolutions per minute made by the engines,

$A \times (d + c) \times 2N$ = cubic inches of steam, at pressure p, used per minute; and the quantity of steam used per hour at the absolute pressure of p pounds per square inch will be

$$= A \times (d + c) \times 2N \times 60, \text{ cubic inches}$$
$$= \frac{A \times (d + c) \times 2N \times 60}{1728}, \text{ cubic feet}$$
$$= \frac{A \times (d + c) \times N}{14{\cdot}4}, \text{ cubic feet.}$$

The volume in cubic feet of one pound of steam at an absolute pressure of p pounds per square inch may be calculated from the formula

$$v = 0{\cdot}41 + \frac{389}{p + 0{\cdot}35}.$$

If, therefore, we divide the total number of cubic feet of steam used at pressure p by the volume of one pound thus calculated, we shall get the number of pounds of steam, and therefore of water, used per hour as shown by

the indicator diagram; and this, divided by the indicated horse-power developed, will give the weight of water used per indicated horse-power per hour:

Or, number of *cubic feet* of steam at absolute pressure p used per hour

$$= \frac{A \times (d + c)}{14 \cdot 4} \times N;$$

therefore, the number of *pounds* of steam or water used per hour

$$= \frac{A \times (d + c)}{14 \cdot 4} \times \frac{N}{v},$$

and the number of *pounds of water used per indicated horse-power per hour*

$$= \frac{A \times (d + c)}{14 \cdot 4} \times \frac{N}{v \times \text{I.H.P.}}$$

For the same cylinder the first term is evidently a constant, which might be calculated at first to facilitate the subsequent calculations.

If a table of specific volumes of steam be at hand, the value of v should be taken from it, in preference to calculation from the approximate formula given on p. 507.

The amount of water used may also be calculated approximately as follows:—After estimating as before the number of cubic feet of steam used at pressure p, multiply this quantity by p and divide it by 14·7 to find the equivalent volume of steam at the atmospheric pressure. This divided by 1700 will give the number of cubic feet of water required to produce the steam, 1700 being the *relative volume* of steam at atmospheric pressure. The weight of one cubic foot of fresh water is 62·5 pounds, so that if the number of cubic feet of water be multiplied by 62·5, the product will be the number of pounds of water used per hour. This product divided by the indicated horse-power will be the quantity of water

used per indicated horse-power per hour, as shown by the indicator diagrams.

Putting this in algebraical language, we have:

If D = number of cubic feet of steam used per hour, at absolute pressure p,

$$D \times \frac{p}{14 \cdot 7} = \text{equivalent number of cubic feet of steam at atmospheric pressure.}$$

$$\frac{D \times p}{14 \cdot 7 \times 1700} = \text{number of } \textit{cubic feet} \text{ of water used per hour.}$$

$$\frac{D \times p \times 62 \cdot 5}{14 \cdot 7 \times 1700} = \text{number of } \textit{pounds} \text{ of water used per hour.}$$

$$\frac{D \times p \times 62 \cdot 5}{14 \cdot 7 \times 1700 \times \text{I.H.P.}} = \cdot 0025 \frac{D \times p}{\text{I.H.P.}} = \text{number of}$$
pounds of water used per I.H.P. per hour.

Compound-engine diagrams.—Indicator diagrams taken from compound engines are usually on different scales for the two cylinders, in consequence of the difference in the steam pressures. In general practice with Richards' indicator, when the boiler pressure exceeds 100 pounds per square inch, the scale for the high-pressure diagrams is 48 pounds, and for pressures below 100 pounds 36 pounds per inch. The scale for the low pressure diagrams is usually 12 or 15 pounds per inch. In calculating the indicated horse-power of the engines the mean pressure must be obtained from each diagram, and the horse-power of the high- and low-pressure cylinders calculated separately, and added together.

It was pointed out on page 338 that the total expansion of steam in the two cylinders of a compound engine was the same as if it had been carried out entirely in the low-pressure cylinder only. In order to ascertain from the indicator diagrams the efficiency of the expansion, the diagrams from the high- and low-pressure cylinders should

be reduced to the same scale and combined, the length of the high-pressure diagram being arranged to bear the same proportion to that of the low that the volume of the high-pressure cylinder does to that of the low-pressure cylinder. Copies of the indicator diagrams taken from the high- and low-pressure cylinders respectively of a compound engine

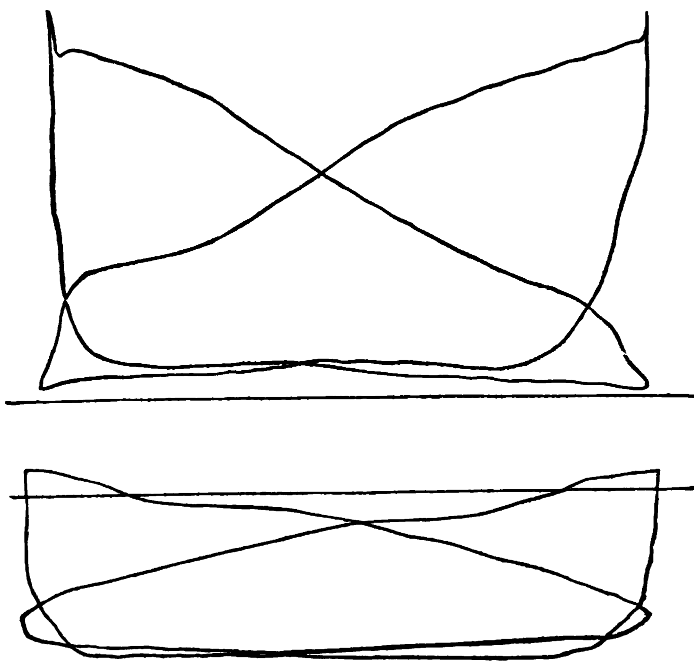


FIG. 214.

with cranks at right angles to each other are shown in Fig. 214. The diameter of the high-pressure cylinder is 55¾ inches and of the low-pressure cylinder 98 inches, stroke of both pistons 33 inches; so that the ratio of the cylinders is 3·09 to 1.

In order to draw the combined diagram, the diagrams from the high- and low-pressure cylinders should be

divided by ordinates drawn across them at equal distances from each other. A base line O P, Fig. 215, should be drawn to represent the zero line, the length of O P being taken to represent the volume of the low-pressure cylinder. O P is then divided at the point Q in such a way that O Q is to O P as the volume of the high- is to that of the low-pressure cylinder. O Q will therefore be the extreme length of the high-pressure and O P of the low-pressure figure on the combined diagram.

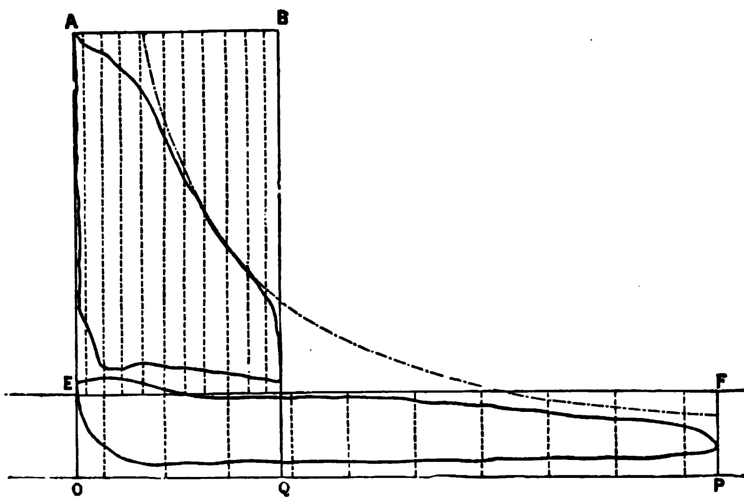


FIG. 215.

Set up O A to represent the initial absolute pressure of the steam to a convenient scale, and draw E F at the proper height to represent the atmospheric line. The lengths, O Q and O P respectively, should be divided into the same number of equal parts as the original diagrams, and ordinates drawn at the points of division.

The several ordinates of the respective indicator diagrams must then be carefully measured, transferred, on the proper scale, to the corresponding ordinate on the

combined diagram, Fig 215, and curves drawn through the ends of the ordinates. It will be seen that the two diagrams fit fairly well into each other and form a tolerably continuous expansion curve. There is, however, a space between the two diagrams, which represents the loss due to the resistance of the passages between the two cylinders, and from sudden expansion on admission to the

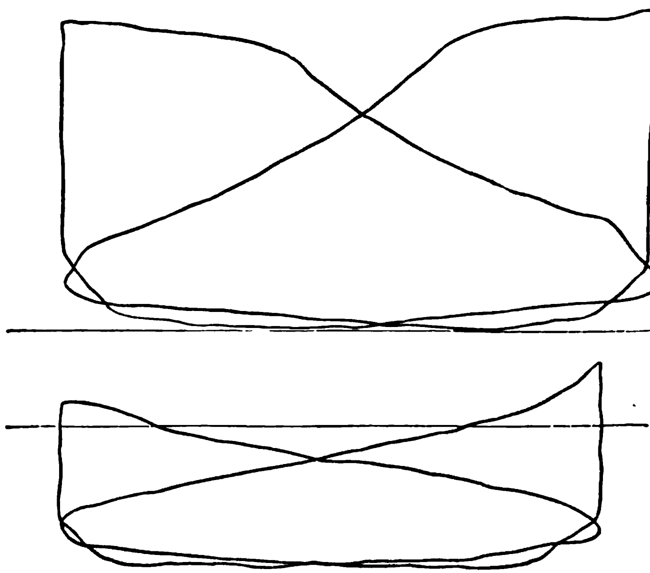


FIG. 216.

intermediate receiver. The combining of indicator diagrams from the cylinders of compound engines in the manner shown, will prove a useful and instructive exercise, and the student will learn many practical lessons from its application.

Specimens of the high- and low-pressure diagrams from the cylinders of the type shown in Fig. 159 are given in Fig. 216. In these engines the expansion is continuous, the back pressure in the high- and the forward

pressure in the low-pressure cylinders at any point of the stroke being the same, excepting the slight loss from resistance of passages. The back pressure line of the high-pressure diagram and the expansion curve of the low are practically identical, and if the two figures are reduced to the same scale of pressures, the low pressure will fit exactly under the high-pressure figure, as shown in Fig. 217.

The subject of indicator diagrams from compound engines is dealt with more fully in the Appendix, but it will be useful to point out a simple method of roughly

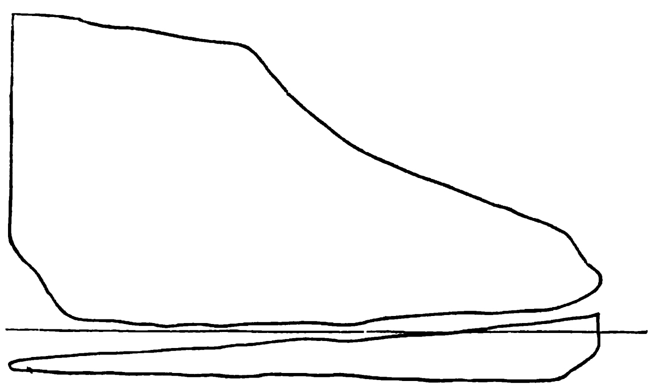


FIG. 217.

approximating to the diagrams that might be expected from a given compound engine working under certain conditions.

In the first place, draw the theoretical diagram, A B C D E, Fig. 218, representing the action of the steam, on the supposition that the whole of the expansion was performed in the low-pressure cylinder, O P being the zero and F G the atmospheric lines, O A the initial absolute pressure of steam, and $\frac{\text{O P}}{\text{A B}}$ the total apparent rate of expansion in the cylinders, neglecting the clearance. In

drawing the curve, the clearance has been taken equal to one-twelfth the capacity of the low-pressure cylinder, as shown by the dotted line O′ A′. The actual rate of expansion of the steam is therefore given by $\frac{O' P}{A' B}$. Let O Q represent the volume of the high-pressure cylinder on the same scale that O P represents that of the low. Draw Q H vertically, cutting the curve at the point H.

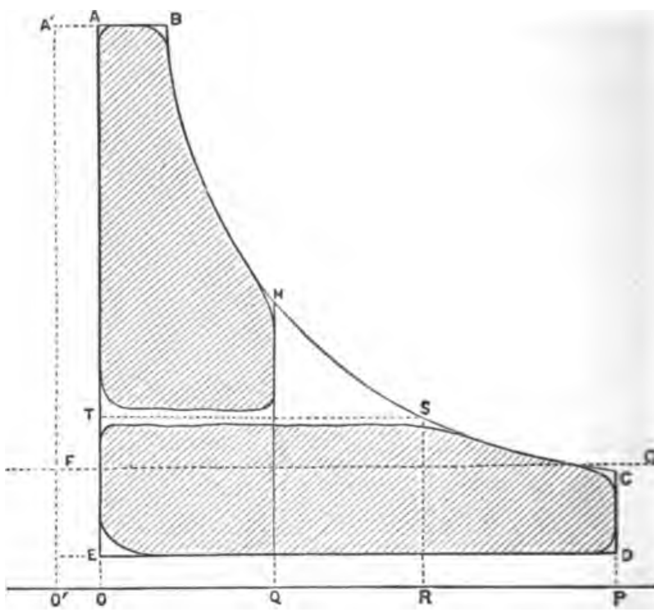


FIG. 218.

The next point to determine is the position of the line dividing the two diagrams. This is found from the point of cut-off in the low-pressure cylinder. Suppose this to be at the point R. Draw the ordinate R S to cut the curve at the point S; then the horizontal line S T, drawn through S, will be the division line between the high- and low-pressure figures, approximately. By rounding off

the corners of the diagrams as shown in Fig. 218, and allowing for the resistance of the passages between the cylinders, say about one pound on each side of the division line S T, we get approximate indicator diagrams, which will at all events be sufficiently accurate for most practical purposes.

It will be seen that the earlier the cut-off in the low-pressure cylinder, the higher will its initial pressure be; so that by increasing the rate of expansion in this cylinder, the work done in it will be increased, and that in the high-pressure cylinder decreased. If the ratio of expansion in the low-pressure cylinder be the same as the ratio of the cylinders themselves, the initial pressure in the low-pressure cylinder will be equal to the pressure at the end of the expansion in the high-pressure cylinder, and there will be no sudden fall of pressure on admission to the reservoir. By means of this diagram the effect of an expansion-valve on the low-pressure cylinder, in regulating the work done by the respective cylinders, may be readily ascertained.

CHAPTER XXV.

PUMPING, WATERTIGHT, AND FIRE ARRANGEMENTS.

THE facilities for pumping out water from steamships, in case of a hole being made in the skin below the water-line, are immensely superior to anything that is possible in sailing-vessels, in which the whole of the work has to be performed by manual labour. A statement of the fittings usually adopted in the ships of the Royal Navy, and of the precautions necessary in order to insure or increase their efficiency, will therefore be useful, and we propose to devote some space to the consideration of this most important subject. We shall not enter into the question of the hand-pumping arrangements, which are practically the same in all ships, whether steam or sailing, but will confine our remarks to the special arrangements which are, or may be, fitted to all ships possessing steam power.

The pumping arrangements should be of such a nature that they may be used whilst the main propelling engines of the ship are at work; so that the ship may be able either to proceed to the nearest port in case of accident, whilst the pumps keep the water from rising too high, or, in extreme cases, that the ship may be kept afloat long enough to be run into shallow water or conveniently beached, to enable the lives of those on board to be saved if possible. In calculating the pumping power of the ship under the adverse circumstances supposed, it must

not be assumed that the engines could be worked at full power, which would give fallacious and misleading results. It is probable that under these conditions the engines could not be expected to work at more than about half power.

Main engine bilge-pumps.—Until recently the majority of steamships have been fitted with bilge-pumps worked direct by the main engines, as described at page 296. These pumps are not fitted so much to provide for extraordinary leaks, as to clear the bilges of the water that drains into them from pipes, bearings, etc., and other ordinary leakages from various causes. They are, however, usually much larger than is absolutely required for these purposes alone, even when the ordinary leakages are comparatively large, and their action should be taken into account in calculating the pumping power of the ship. The plungers of these pumps are always working with the main engines, so that it would not be desirable to make them as large as might otherwise be the case; for it would be difficult to keep the plungers and barrels sufficiently lubricated to prevent overheating; and a considerable addition would be made to the constant friction of the engines, which is already quite large enough in most cases. Larger pumps would also necessitate larger discharge pipes, valves, etc., which would be in any case inconvenient, and often impossible. In recent ships, separate bilge pumping engines have been fitted instead of pumps worked by the main engines.

For sufficient pumping power it is necessary to provide additional pumps, and by examination we shall see how readily a very large amount of steam power can be rendered available, with comparatively little trouble or expense, in ordinary steamships.

Bilge injection.—This fitting has been already described in Chapter XVII., and has the effect of practically

turning the air-pump into a bilge-pump, as all the water used for condensing the steam would be taken direct from the bilge and pumped overboard through the main discharge valve. This is fitted to all jet injection condensers, and affords a ready means of getting rid of a large quantity of water in case of a leak. It is also fitted to many surface condensers, and as this would not in any way interfere with the operation of the circulating-pump when that was used for bilge purposes, both the air- and circulating-pumps might be utilised for clearing the ship of water in case of necessity. As pointed out at page 307, when the air-pump is vertical and single-acting a bilge-suction is often led to the pump in lieu of a bilge injection. In recent ships the circulating-pumps for the surface condenser have been duplicated and the bilge injection fittings dispensed with.

Circulating-pump.—The pumps for circulating water through the engine condensers may be considered as the main reliance in case of danger. These pumps are fitted with bilge- as well as sea-suctions, to be used in case of need, the water being drawn from the bilge instead of the sea for circulating through or around the tubes. Additional pipes to discharge the water directly overboard without passing through the condensers are fitted in all recent ships.

If the pumps be reciprocating and worked by the main engines, their efficiency and power would be reduced in the same ratio as the revolutions of the engines, which could only be worked at reduced power under the circumstances. If the circulating-pumps be worked by separate engines, the above limitation would not exist to nearly the same extent.

Circulating-pumps, when worked by separate engines, are usually centrifugal, and the efficiency of these pumps is not so well known as that of ordinary reciprocating-pumps, but there is sufficient evidence to prove that for

the small lift required to pump out a ship, the quantity of water that can be thrown by a comparatively small centrifugal pump is very great. The quantity of water discharged by a centrifugal pump depends mainly on the velocity of its periphery; but as a rule, it would appear that for a given peripheral velocity large pumps driven at a comparatively low number of revolutions are more efficient than smaller pumps revolving more rapidly.

In the Royal Navy the efficiency of the circulating-pumps for bilge purposes is always tested prior to the receipt of the machinery. In the armour-clad ships of the 'Admiral' class four centrifugal pumps, each 4 feet 3 inches diameter, are fitted for the engine-condensers, and these are capable of discharging at least 3,200 tons of water per hour from the bilges when worked at a speed not exceeding 250 revolutions per minute.

Precautions have to be taken to insure the efficient working of centrifugal pumps for the purpose of lifting water. For ordinary work in ships, their only office is to circulate the water, both the inlet and outlet orifices being below the surface. They cannot in all cases be relied on to lift the water from any considerable depth, and it is very desirable that they should be placed as low down in the ship as possible, which is the most simple way of insuring that they will draw readily in the excitement consequent on the ship making water fast. It is also necessary that they should be fully charged with water at starting, and kept fully supplied when at work, for any air getting into the suction-pipes soon stops the action of the pumps. Self-acting non-return valves should therefore be placed at the bottoms of the suction-pipes, close to the bilge, so that the whole length of the suction-pipes may be filled with water before starting the pumps to draw from the bilge. The suction-pipe should also be of sufficient area to keep the pump fully supplied with water, without requiring

too high a velocity in the pipe. It is probable that in some cases the inefficient action of these pumps for bilge purposes may have been due to the smallness of the supply pipe, the water not being able to enter the pump sufficiently fast to keep it fully charged. The bilge suction pipe should be at least as large as the delivery in order to insure the most efficient results.

Though the pump itself should be close to the bilge, the engine for working it should be at a high level—if possible above the load water-line—so as to be out of the reach of the water in case of its rising rapidly, as the only chance of pumping overboard with a centrifugal pump is to maintain a high speed of revolution.

This arrangement has been carried out in the 'Inflexible,' 'Nelson,' and several other ships, the pump being placed in a horizontal position in the bilge, and worked by a vertical shaft carried to a considerable height, and attached to the crank-shaft of the engine, which revolves in a horizontal plane. No engine can work long with water surrounding it, turning the cylinders into condensers and stopping the engines. The arrangement above referred to appears to be very suitable, the pump being in the position in which it acts most efficiently, whilst the engine is fixed on a platform at such a height that the main boilers and engines of the ship would be almost entirely submerged before the water could reach the cylinders of the circulating-engine, so that the centrifugal pump could be kept at work until the last possible moment.

Another point that should be kept in view in designing these pumps is the provision of large engine power; for in the case of a serious leak, when the water rises and gathers on the fires, or prevents the firemen from working in the stokeholds, the pressure of steam will drop, and would probably before long become even less than the atmospheric pressure. For this reason, as well as for the

sake of economy, the exhaust steam from the circulating engines should always be taken to the main engine or auxiliary condenser, and the air-pumps kept working, if possible, to maintain the vacuum.

The valves for changing the suction of the centrifugal pumps from the sea to the bilge are arranged to be worked from the starting platform, and to enable this to be done quickly in case of need, the valves in the sea and bilge suction-pipes are coupled together so that they may be worked by a single lever.

Several instances have been recorded in which ships that have been seriously injured have been kept afloat long enough to run to harbour, or at all events to enable the passengers and crew to be saved, by the application of the circulating-pumps to pump the water overboard.

Hand-pump.—This is a pump, still sometimes fitted, which is arranged to be worked either by hand or by an eccentric sheave or pin on the engine-shaft. It is generally fitted for several purposes, one of which is for pumping out the bilge, so that it may be added to the available steam pumping power. In recent ships the hand-pump is fitted principally for testing purposes, and is not worked by the engines.

The arrangements just enumerated comprise all the pumps that are connected with the main propelling machinery of the ship. In ships of the Royal Navy, there is, in addition, considerable auxiliary pumping power.

Fire-engine.—First in importance is the fire-engine, which is always fitted with a bilge, as well as a sea-suction, so that it may be utilised for pumping out the ship if necessary, as well as for extinguishing fires. This engine is generally placed at a considerable height in the ship, and the regulating suction-valves to the pumps, both from the sea and bilge, should be close to the engine, so that

they may be always readily accessible. Self-acting non-return valves should be fitted at the lower ends of the bilge suction-pipes, to prevent the possibility of water passing into the ship when the sea-suction is open. These engines are sometimes fitted with separate condensers, so that they can be worked with reduced steam pressure if necessary. When the engine is above the water-line the injection water may be obtained from the pump discharge. In recent war-ships auxiliary surface-condensers are fitted, into which the steam from all the auxiliary engines of the ship exhausts. This condenser is fitted with a separate circulating pump and a small air-pump to discharge the condensed water into the feed-tanks.

Second auxiliary bilge engine.—There is also at least one smaller auxiliary pumping-engine fitted in ships of the Royal Navy, which is used for ordinary bilge purposes, washing decks, etc., and which may therefore be added to the pumping power of the ship in case of a leak.

The foregoing constitute what may be called the ordinary steam pumping arrangements of the ship, and we now come to the special pumping appliances that have been fitted in some cases.

Special fire-engine.—In some ships special high-pressure fire-engines, with independent boilers, similar to those made by Messrs. Merryweather, Shand & Mason, etc., for land purposes, have been fixed on the main deck, or in some other convenient position above the water-line, the pumps being arranged to draw either from the sea or the bilge. The boilers supplied with these engines are specially fitted for the purpose of raising steam rapidly, say in from 10 to 15 minutes, the water being cold on starting; and as the pumps attached are of a powerful character, they would be valuable in case of necessity, either for pumping out the ship or for extinguishing fire.

There is, however, one serious drawback to their use on

shipboard, namely, the necessity of supplying the boiler with fresh water only, for feed purposes. The period of efficient action of these pumps would therefore be limited by the quantity of fresh water in the ship available. A fresh-water supply-tank is usually fitted near the boiler, and the tanks containing the drinking-water for the use of the ship's company might be connected to this feed-tank, so that in case of emergency the whole of the fresh water in the ship could be made readily available for the purpose of supplying the boiler and keeping the engine and pumps at work as long as possible. Salt-water feed would be dangerous, as the ends of the water-tubes, being exposed to the fierce heat of the fire, would get choked and burnt.

Friedmann's bilge-ejector.—This apparatus is a modification of the Giffard's injector used for feeding boilers. In Giffard's injector the water is forced into the boiler by the action of the steam rushing through a conical orifice in

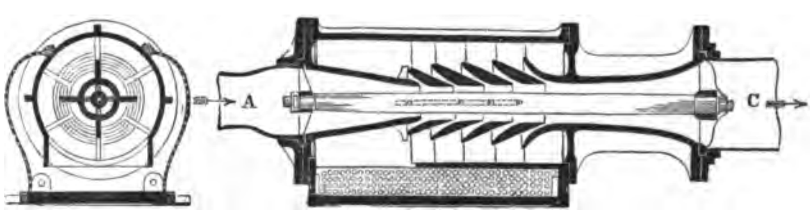


FIG. 219.

the end of the steam-pipe, over the open end of the water-supply pipe, the water being carried in with the steam. In Friedmann's ejector the number of nozzles is increased, so as to give the steam several suction-orifices instead of one. The arrangement is shown in Fig. 219. The steam from the boiler is conducted through the pipe A to a tuyere in the pump, about one-half the diameter of the steam-pipe, and then passes successively through a series

of intermediate tuyeres, through which the water is drawn from the hold, and expelled from the ship through the discharge pipe C. This apparatus occupies very little space, and though it is difficult to obtain reliable data as to the quantity of water it is capable of blowing out of the ship when in full action, there can be little doubt that it would be very great.

From the nature of the arrangement it is clear that it would consume a large quantity of steam, and would not be economical for general use; but considerations of economy necessarily disappear when the safety of the ship is in question, and the object of the fitting is to provide the means of rapidly getting rid of a large quantity of water when the ship is in danger. It lends itself more especially to the case of high-pressure steam, for the higher the velocity of the steam through the tuyeres the greater will be the efficiency of the apparatus.

Several of the armour-clad ships of the Royal Navy are fitted with Friedmann's ejectors. The diameter of the steam supply pipe is about 5 inches, and of the tuyeres 2½ inches. The water discharge pipes, which have simple hinged non-return valves on the ship's sides, are about 8 inches in diameter. The suction-pipes are led to tanks in the bilges, and the entrances are carefully protected with gratings to prevent chips, etc., being carried in to choke the apparatus. It is not probable that if these ejectors were at work there would be much, if any, steam to spare for the main engines, or for other auxiliary pumping machinery. They would, therefore, only be used in cases of extreme urgency, when it is only possible to keep the ship afloat for sufficient time for assistance to arrive.

Small bilge ejectors, with a single orifice, are fitted to all the steam launches, pinnaces, etc. in the Service, and have been found by experience to be most efficient for

clearing the boats of water. The steam-pipe is $\frac{1}{2}$-inch diameter, with a $\frac{1}{4}$-inch orifice, and the discharge-pipe is $\frac{7}{8}$-inch diameter. This small ejector has been found by experiment to be sufficiently powerful to force water out of the bilges faster than it would flow in when the plug, about one inch diameter, in the bottom of the boat was removed. This simple illustration may perhaps give some idea of the pumping power that might be expected from Friedmann's improved apparatus.

Pulsometer.—This is a peculiar pumping arrangement which acts by the direct pressure of the steam on the water. It has been fitted in several steamships for various purposes, and appears to have given satisfaction. A section of one is shown in Fig. 220, and its action is stated to be as follows:

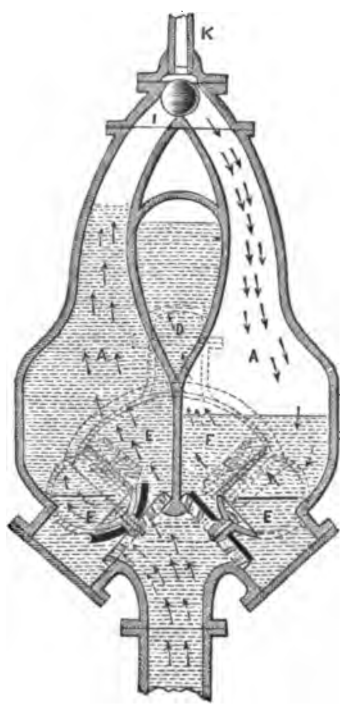


FIG. 220.

The pump being filled with water, either by pouring water through the opening in the chamber or by drawing the charge, is ready for work. Steam being admitted through the steam-pipe K, by opening to a small extent the stop-valve, passes down that side of the steam neck which is left open to it by the position of the steam-ball, and presses upon the small surface of water in the chamber A which is exposed to it, *depressing it without any agitation*, and, consequently,

with but slight condensation, and driving it through the discharge opening F and valve into the rising main D. The moment that the level of the water is as low as the orifice which leads to the discharge, the steam blows through with a certain amount of violence, and being brought into intimate contact with the water, *an instantaneous condensation takes place*, and a vacuum is in consequence so rapidly formed in the just-emptied chamber, that the steam-ball I is pulled over into the seat opposite to that which it had occupied during the emptying of the chamber, closing its upper orifice and preventing the further admission of steam, allowing the vacuum to be completed; water rushes in immediately through the suction-pipe, lifting the inlet-valve E, and rapidly fills the chamber A again. Matters are now in exactly the same state in the second chamber as they were in the first when our description commenced, and the same results ensue. From this description it will be seen that there are no glands, no stuffing-boxes, no eccentrics, no packing, no slide-valve, and in fact no working parts except the valves, which can be easily and cheaply renewed when necessary.

This pump is noiseless in action, and so simple in construction that it cannot well get out of order, and it would probably keep at work continuously with very little cost for repairs. From the method of its action, however, it is not probable that it can be economical in the expenditure of steam. It bears a very close resemblance in principle to Savery's engine (which was patented in 1698), but improved and simplified in the details of construction. The makers of this instrument state that a pulsometer fitted with a 2-inch steam-pipe, 10-inch suction-pipe, and 8-inch discharge-pipe, will discharge about 290 tons of water per hour.

We have now mentioned all the steam pumping appliances supplied to ships most perfectly equipped in this respect, and though these are most valuable, we think

most ships would be in a very sorry plight in case of serious damage below the water-line if they had only the pumping arrangements to rely on.

The quantity of water that would flow into a ship through a hole in her bottom may be calculated approximately as follows:

Let H = depth of hole below the water-line in feet,

and A = area of hole in square inches.

Then, if V = velocity of flow into the ship in feet per second,

$$V = \sqrt{2\,g\,H}.$$

Where $g = 32{\cdot}2$ = accelerating force of gravity.

Therefore

$$V = 8\sqrt{H} \text{ nearly.}$$

The number of cubic feet of water that would flow into the ship per second is therefore equal to

$$\frac{A \times 8\sqrt{H}}{144} = \frac{A \times \sqrt{H}}{18}.$$

For example, supposing a hole 12 inches in diameter to be made in the skin of a ship, 16 feet below the surface of the water. The area of this hole is equal to 113 square inches, so that the rate at which the water would begin to flow into the ship would be $= \frac{113 \times \sqrt{16}}{18} = 25$ cubic feet per second, or 90,000 cubic feet per hour. This divided by 35 will give the number of tons per hour; so that a hole 12 inches in diameter, 16 feet below the water, would be capable of admitting into the ship $\frac{90{,}000}{35} = 2{,}570$ tons of water per hour. It is easily seen by a calculation of this sort how comparatively small a hole in the bottom of a ship would be sufficient to absorb all the available pumping power that could be provided.

The following constants, which are frequently required, are given to facilitate calculation :

One gallon = 277·274 cubic inches.
= 10 pounds (fresh water).

One cubic foot = 6·25 gallons.
= 62·5 pounds (fresh water).
= 64 pounds (sea water).

35 cubic feet of sea-water weigh one ton.

35·84 „ fresh water „

The pumping arrangements connected with the main propelling machinery should be made, in all ships, as powerful and efficient as circumstances will allow. Possibly, the auxiliary pumping arrangements enumerated could not be completely adopted in all ships, though it is desirable that they should be to as great an extent as may be practicable. Efficient pumping arrangements for every steamship, if properly considered and fitted, would not involve much additional expense, and would greatly add to the safety of the ship, and keep her afloat after receiving considerable injury to the hull below the water-line for sufficient time to enable the ship to reach a port or to run into shallow water.

They can, however, when most complete and efficient, only be regarded as an auxiliary in cases where the ship sustains considerable damage below the water-line, such as from collision, etc.; and the only complete safeguard is the division of the hull into as many watertight compartments as is possible, consistent with the requirements of the ship, so that the effects of any injury may be localised. The pumping arrangements could then probably, in many cases, be utilised to keep the defective compartment sufficiently free from water to enable the damage to be wholly or partially repaired.

Watertight bulkheads.—The watertight compartments are formed by constructing iron or steel watertight bulk-

heads across the ship at certain sections. In some cases there is a further subdivision by longitudinal bulkheads, and in the more recent ships, horizontal iron or steel watertight decks have been fitted. When the vertical bulkheads do not terminate at a watertight deck, it is clear that their upper edges should be carried to a sufficient height above the load water-line to prevent the water flowing over the tops of the bulkheads into the adjoining compartments, even when the ship is at the greater immersion due to the injured compartments being full of water. The height to which the bulkheads are carried above the load water-line should be proportioned to the volumes of the respective compartments in order to insure safety without unduly increasing the weights.

Watertight doors.—The communication between the various compartments is maintained by means of sliding watertight doors, sketches of which are given in Figs. 221 and 222. When the door is worked in a vertical direction, it is raised and lowered by means of a screw, as shown in Fig. 221. This screw should be double- or treble-threaded, with a coarse pitch, in order to enable the door to be closed with sufficient rapidity when necessary. When the watertight door slides horizontally, it is worked by means of racks and pinions, the racks being fixed on the back of the door itself, as shown in Fig. 222. The gear for working these doors should be on the main or upper deck, above the water-line, and in positions readily accessible in case of emergency. Means should also be provided for working the doors from below when required. Wedges are fitted on the doors and at the ends of the guides, so that when the door is shut it may be pressed tightly against the inner face of the guide, to prevent the passage of water. To promote uniformity and prevent mistake, the gear for working watertight doors should be fitted in such a manner that the door may be closed by a right-

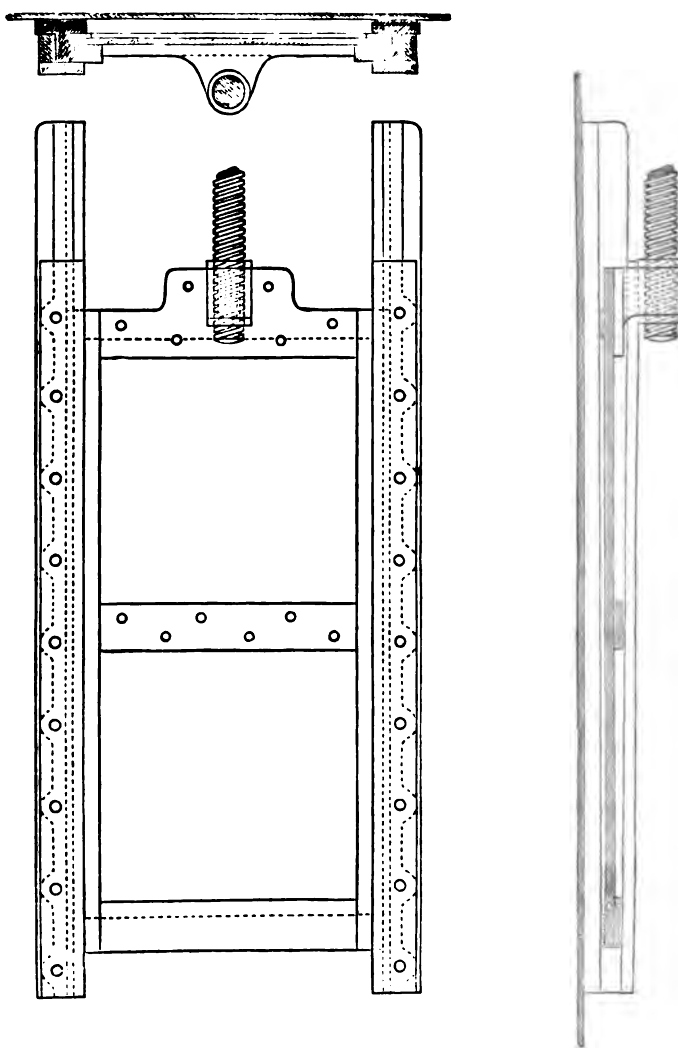


FIG. 221.

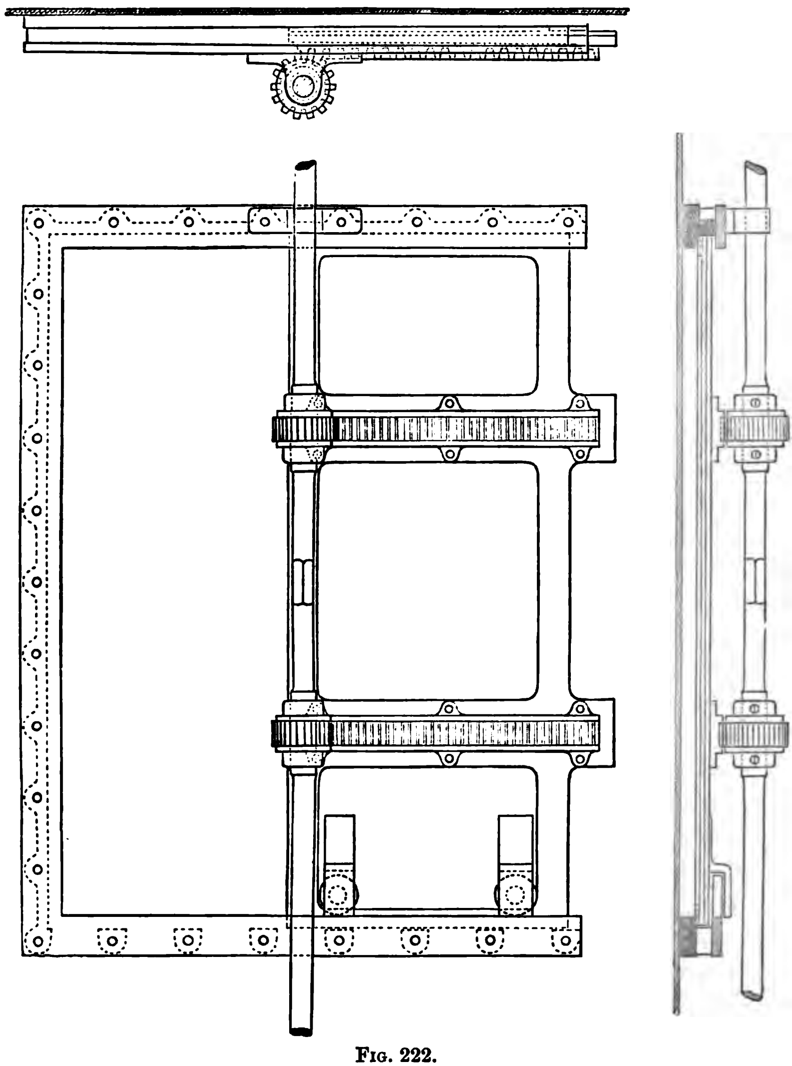


FIG. 222.

handed motion. Every precaution should be taken to keep the guide grooves clear and the gear in good working order, so that there may be no difficulty in closing the doors in case of necessity, when any mistake would be not only inconvenient but fatal. When the doors are open, the bottom grooves, which are the most liable to become choked, are usually covered with plates or sills, fitted to open out automatically by the closing of the doors. The doors in watertight bulkheads should be as few as possible in number, and placed at as high a level as circumstances will allow, in order to increase the efficiency of the watertight arrangements.

Sluice-valves.—Small sluice-valves are usually fitted in the lower parts of the watertight bulkheads to allow the water to be drained from one compartment to another if required. These valves should be arranged to shut with a right-hand motion, and the rods, etc. for working them should be carried to the same height, and as nearly as possible to the same positions, as the gear for working the watertight doors.

Double bottom.—The safety of many ships in the Royal Navy, especially the armour-clad ships, is also very much increased by the construction of what is generally known as the double bottom. This consists of an inner watertight skin, at some distance from the outer skin, extending for about ⅔rds to ¾ths of the total length of the ship. The distance between the inner and outer skins at the bottom of the ship is generally about three or four feet; and above the turn of the bilge, the inner skin is continued by vertical bulkheads carried up above the waterline, and forming what are called the wing passages. The double bottom and wing passages are divided into many small compartments by longitudinal and transverse bulkheads. To show the extent to which subdivision of ships of the Royal Navy has been carried, we may state

that in the 'Inflexible' there are 89 compartments in the hold space, and 46 in the double bottom and wings, making a total of 135 watertight compartments in the ship.

Man-holes.—In order to enable the surfaces of the plates in the double bottoms and wing passages, etc., to be examined, cleaned, and painted, when necessary, suitable man-holes are made in the inner skin of the ship and in the bulkheads dividing the several compartments. The man-hole doors are screwed down over the holes, with red-lead putty between the faces, so as to make perfectly watertight joints.

Considerable damage might be done to the outer skin of a ship fitted with a double bottom, without endangering the safety of the ship; for unless the inner skin be broken no water can enter the hold. The compartments in the double bottom are also so small that the filling of several of them would have but comparatively little effect on the immersion or the trim of the ship.

Water ballast.—In some cases the double-bottom compartments have been utilised for the carrying of water ballast, and in this way they have proved useful as the ship became lighter from the consumption of coals, stores, &c.

Double-bottom suction-pipe.—One or more of the steam pumping-engines in the ship should be fitted with a suction-pipe having branches leading into each of the several compartments of the double bottom, for the purpose of pumping them out. This suction-pipe may also be utilised for filling the compartments when required, by taking out the non-return valve at the pump and opening the sea-suction. There should be a separate suction-cock or valve on the branch pipe leading to each compartment.

Main drain-pipe.— To facilitate the pumping out of

the hold of a ship divided into several compartments, a large drain-pipe is often carried through the double bottom, with small branch connections leading to the several compartments of the hold. At the ends of these connections drain-valves are fitted, which may be opened or closed as required, by rods carried to a high level. Instead, therefore, of leading suction-pipes from the pumping machinery to each compartment of the ship, it is only necessary to connect one suction-pipe from each of the pumping-engines to a convenient part of the main drain-pipe, with which any particular compartment may be connected by opening the drain-valves. This more particularly applies to the compartments outside the engine and boiler rooms; for although these may also be drained into the main drain-pipe, it is generally desirable to fit, in addition, an independent suction-pipe from the principal auxiliary steam-pumps to the engine-room or stokehold bilge. Though for convenience the main drain-pipe is carried through the double bottom, it is not in connection with it, and is only fitted for the drainage from the compartments in the hold of the ship.

Fire arrangements.—We have now described the pumping and watertight appliances necessary to promote safety in case of the ship springing a serious leak, and pass on to consider the precautions necessary to provide against the danger of fire.

Flooding-cocks.—To the most dangerous parts of the ship, such as the magazines, shell and spirit rooms, etc., flooding-pipes are led, connected with the sea by Kingston valves and sea-cocks in the ordinary manner; so that in case of fire, the compartment might be flooded with water from the sea, if considered necessary. Special precautions have to be taken to prevent these cocks being tampered with; and, as a rule, to insure safety, there is between the ordinary sea-cock and the compartment an additional cock

or valve which is locked and can only be opened by the officer entrusted with the key.

Fire-main.—All the auxiliary steam-pumps described in the earlier part of this chapter, that may be used for pumping out the ship in case of a leak, are also fitted with sea-suctions, and may be caused to deliver the water into a pipe called the fire-main, which is carried fore and aft in the ship, with branches leading to different parts as required. At various points in the fire-main, delivery-cocks, with suitable screwed nozzles, are fitted, to which hoses may be connected to direct the water on any required spot. Non-return valves should be fitted at the junctions of the delivery pipes from the several pumping-engines with the fire-main; and the pumps should have sufficient power to produce a pressure in the main at least equal to throwing the water from the nozzles of the delivery hoses over the masthead.

CHAPTER XXVI.

AUXILIARY MACHINERY AND FITTINGS.

In modern war-ships the auxiliary machinery and fittings occupy a position scarcely, if at all, inferior to that of the main propelling-engines themselves. In fact, so far as the anxieties and responsibilities of the engineer officers of such ships are concerned, they will be found to be at least as great outside the engine-room as in it, the auxiliary engines being so numerous and the machinery and gear connected with them of such varied description and construction. In such a ship as the 'Inflexible,' for example, there would be between 40 and 50 different engines for various purposes, and a large number of hydraulic lifts and other machines, in addition to the main engines of the ship. The ship herself may in point of fact be regarded as a huge floating war-machine rather than a ship properly so called, and all the necessary operations for working, steering, and fighting her are performed by steam alone, without the intervention of manual labour; so that her efficiency will depend entirely on the condition of the machinery department.

A simple enumeration of the various kinds of work on board ships that are now done by steam power would be sufficient to show the importance of this part of the duty of a naval engineer, and it is necessary that a little space should be devoted to the description of the more important auxiliary machinery and gear now fitted in war-ships. In

addition to the main propelling-engines, steam power is used for ventilating the ship, supplying air to the furnaces of the boilers, weighing the anchor, steering, pumping, working turrets, loading, training, and working the guns, compressing air for charging and launching torpedoes, putting torpedo and other boats into and lifting them out of the water, distilling, producing electric currents for lighting purposes, special or general, signalling, etc.

The design and arrangement of the auxiliary machinery and fittings of the ship should be regarded and treated as a whole, in order that uniformity and simplicity may be introduced to as great an extent as possible. It is of great importance, in war-ships especially, that complexity, which must in any case be considerable, should be reduced to a minimum, in order to insure efficiency. In cases where the auxiliary machinery is supplied by different and independent makers, who may have developed certain specialities, each engine will be an independent item, designed and fitted without reference to any other, so that special study has to be made of each. It is not difficult to make a general arrangement of the whole of the auxiliary machinery and fittings required for any particular ship, and if this system were universally carried out, many simplifications would probably be suggested from time to time in consequence of the special study and attention given to the subject. In a factory or establishment on shore, one engine is usually employed to work the whole of the machinery by means of shafting, etc. This would not be practicable on board ship, but it is possible that a central pumping-engine might be fitted to work the whole of the auxiliary machinery by hydraulic power or compressed air; and this, if conveniently arranged, would doubtless conduce both to economy and efficiency. It is not at all improbable that the development of electrical science may

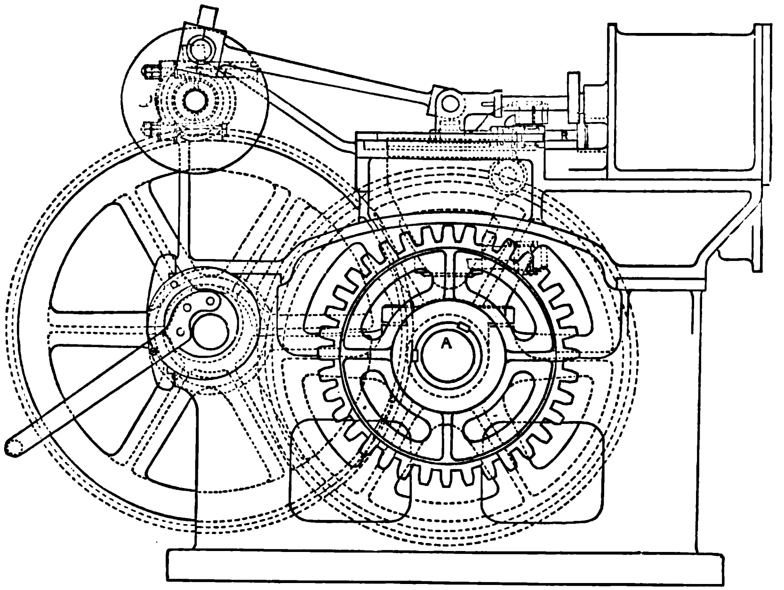
before long render possible the transmission of the power generated in a central dynamo-machine to electric motors for various purposes in different parts of the ship. This would be a great advance, for with the present system, the arrangement of the steam and exhaust pipes, valves, etc., for the several auxiliary engines must necessarily be very complicated in many cases.

The pumping, watertight, and fire arrangements were explained in the previous chapter, and need not be further discussed. We now proceed to describe some of the more important auxiliary machinery and gear fitted for other purposes.

Steering-engine.—Figs. 223 and 224 show an arrangement of steam steering engine, patented by Mr. McFarlane Gray, which has been fitted in a large number of ships, and found in practice to be very efficient. The shaft A, from which the power is transmitted to the tiller, is driven by the engine through a system of toothed gearing, which reduces the speed of revolution to a sufficient extent to obtain the necessary turning moment to work the rudder readily, when the ship is moving through the water at her maximum rate of speed. The details of the arrangement are clearly shown in the diagrams.

The speciality in this engine is the fitting by which it is stopped when the helm is moved to the required angle, the rudder being held in this position until the engine is again moved by the working of the steering-wheel. A sketch of this arrangement to an enlarged scale is shown in Fig. 225. The starting and reversing valve D, Fig. 224, is a differential valve, similar in construction to that described on page 267, and it is actuated by means of shafting and gearing, led from the engine to the steering wheels placed at various parts of the ship. The rod of this reversing valve is marked R on the diagrams, and it will be seen that it is actuated by the rising and falling of the screw S,

FIG. 223.



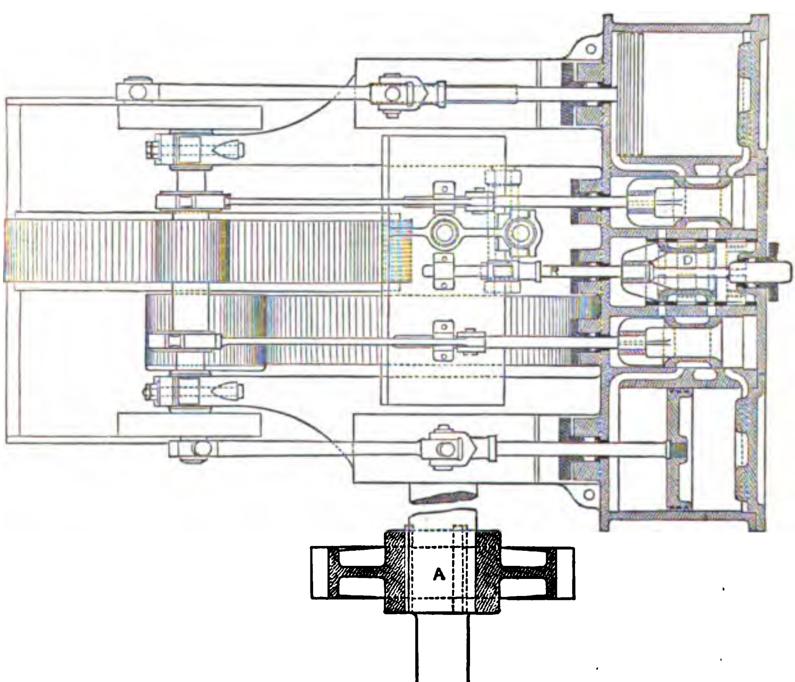


FIG. 224.

through the bell-crank lever working on the fulcrum B, Fig. 225. The screw S is caused to revolve by the action

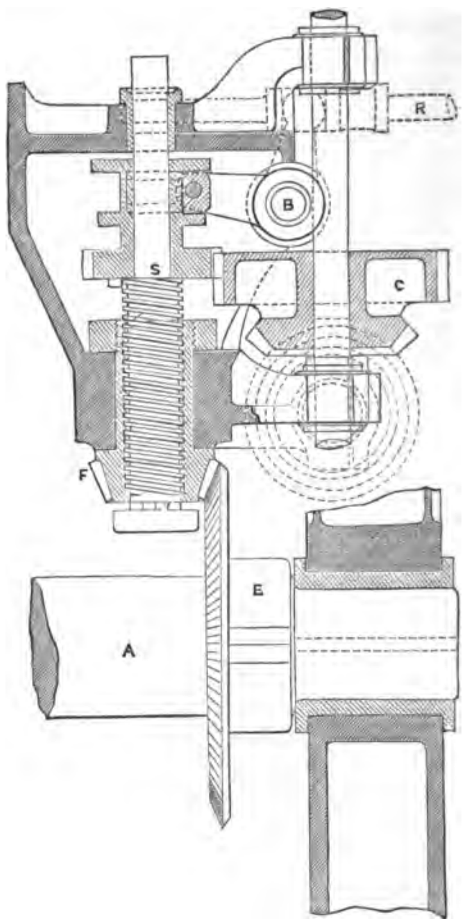


FIG. 225.

of the spur-wheel C, which is worked by means of the shafting and gear connected to the several steering-wheels in the ship. On the shaft A a bevel-wheel E is fixed,

which, as the shaft revolves, works the nut F, and thus causes the screw S to rise or fall according to the direction of rotation. This gear is so arranged that when the engine is started in any direction by the working of the steering-wheel, the action of the wheels E and F tends to move the starting-valve back to its central position, and thus to stop the engine. There are, therefore, two forces acting on the valve: the one from the steering wheel, worked by hand, which moves the valve to the right or left as may be required to start the engine in the direction necessary to put the rudder to starboard or port as desired; and the other from the engine itself, which always tends to replace the valve in its central position and stop the engine. Consequently, the engine only keeps in motion while the steering-wheel is being worked, and as soon as the steersman ceases turning the wheel the steering-engine stops, and the rudder remains fixed in the position to which it has been put, until it is again moved by the further action of the steersman.

There are many other arrangements for producing this result in different types of steering engines, but the essential principle in all is the same, viz. that the engine as it revolves always tends to reverse the action previously produced on the starting-gear by the steering wheel, so that the engine can only be kept working while the steering wheel is being moved, and stops directly the wheel is stopped.

On the pedestals that carry the steam steering wheels *controlling boxes* are fitted which contain safety arrangements or stops to prevent the rudder being placed at too great an angle either to starboard or port, which would strain and endanger the steering gear.

The steering engine sometimes works, through the shaft A, a barrel on which hide ropes are wound, which lead to the end of the tiller as in the ordinary hand steer-

ing gear. This, however, is not suitable when steam power is used, as there is considerable risk of the ropes being broken if the engine be moved quickly; and the holes in the bulkheads of the ship through which the ropes pass cannot be made watertight. More generally, therefore, shafting is carried from the steering engine to the after compartment of the ship, in the tiller-room itself if possible, where it drives, by means of bevel gearing, a vertical

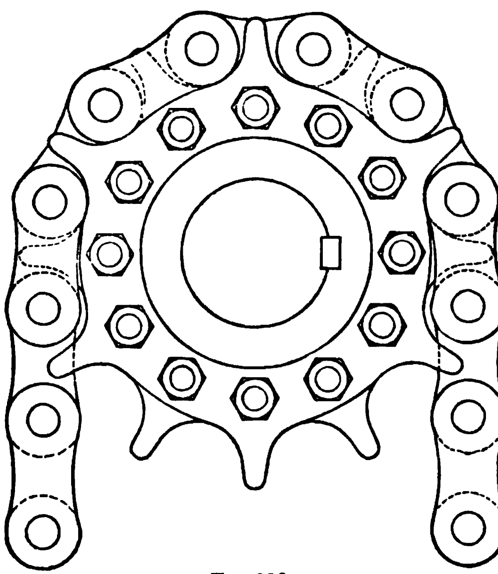


Fig. 226.

shaft carrying a horizontal chain pulley, or sprocket-wheel, which acts on a pitch-chain attached to the tiller. This is shown in Fig. 226.

In some ships, instead of the sprocket-wheel and pitch-chain, a barrel is keyed on the end of the shaft, and carries a steel wire rope which is attached to the tiller in the ordinary manner. This works more smoothly than the pitch-chain. The barrel should be turned and spiral

grooves cut in it to guide the wire rope and prevent it from chafing.

In all gear worked by steam power, the arrangements should be as rigid and mechanical as possible, and the system of shafting and gearing should always be adopted, if practicable, in preference to ropes or chains. The holes in the bulkheads of the ship through which the shafting is carried can be made watertight by suitable stuffing-boxes, so that this plan does not prejudice the watertight arrangements of the ship.

Expansion couplings.— Suitable expansion couplings should be fitted in all great lengths of shafting in the ship, to make provision for the working of the ship, or for alteration of length due to change of temperature.

Hydraulic steering-gear.—In many ships, hydraulic power has been used, instead of steam power, for working the rudder, and various plans have been adopted for utilising water pressure for this purpose.

In the earlier hydraulic gears, the water pressure was simply applied, in lieu of ropes or chains, as the agent for transmitting the power of the hand steering-wheel to the rudder, and thus to reduce the losses from friction, etc. The tiller was acted on by two rams placed athwartships, one on each side, and the steering wheel worked, by means of a screw, a piston in a water cylinder the ends of which were connected by small pipes to the port and starboard rams respectively. When the whole system was filled with water and hermetically sealed, it is clear that by the working of the piston in the hydraulic cylinder, pressure would be transmitted to the rams by the water in the pipes, and the tiller worked as desired. The principal difficulties in this arrangement arise from the leakage of the piston and rams, which tend to make it slow and uncertain in action. The work also is performed by manual labour alone, the only use of the water being to

transmit the power applied to the hand steering wheel, and thus to save the losses resulting from friction, etc., in ordinary mechanical steering gears, when the steering wheel and rudder are at a considerable distance from each other.

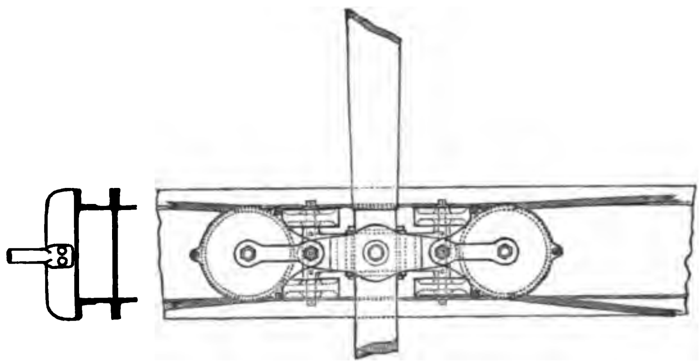
In the more recent hydraulic steering gears manual labour is dispensed with, and a water pressure of about 700 to 800 pounds per square inch is produced in the apparatus by means of a steam pumping-engine. This pressure is kept constant by the action of a special accumulator, or of a spring-loaded piston-valve connected by levers or other gear to the starting-valve of the pumping engine. The tiller is acted on by rams, as in the previous case, and the water pressure is admitted to the rams by a slide-valve, worked by a small steering wheel or lever. On moving the slide-valve the water passes from the pumping-engine through the open port of the valve to one of the driving-rams, and escapes from the other ram, through the exhaust port of the slide-valve, to the suction-tanks from which the pumps draw their supply, so that the same water is constantly used. This is convenient, as it enables glycerine to be added to the water to prevent freezing in cold weather. The rudder itself is also connected to the steering slide-valve, and always tends to close it as it moves from its central position, and thus to reverse the action of the steering wheel, as in the steam steering gear previously described. When, therefore, the steersman stops moving the steering wheel, the rudder replaces the valve in its central position and closes the ports, and consequently remains in the required position until again moved by the opening of the valve ports. The hydraulic rams on the tiller are usually fitted with spring escape-valves to prevent undue straining of the pipes from any sudden increase of pressure resulting from seas striking the rudder, etc.

In several ships that have been fitted with this kind of steering gear, the hydraulic pressure has been utilised for other auxiliary purposes, such as for working cranes, winches, reversing gear for main engines, &c.

The advantages of fitting steam or hydraulic steering gear to dispense with manual labour, especially in the case of large ships, need no discussion. They place the control of the ship practically in the hands of one man, who is, by their aid, enabled to place the rudder hard over to starboard or port, when the ship is steaming at full speed, with little exertion, perfect safety, and in very much less time than would be required with hand gear, even when worked with a large number of men. It is, however, essential that the steering apparatus should be simple in construction and arrangement, and safe and reliable in operation: and much skill, care, and attention are required to be devoted to the design of the whole of the details of the gear to reduce complexity and promote efficiency to as great an extent as possible.

Rapson's slide.—Various methods are adopted for connecting the steering gear to the tiller. One of the most mechanical and efficient is that known as Rapson's slide, which is designed to promote uniformity in the turning moment for all angles of the rudder. A sketch of this arrangement is shown in Fig. 227. The details of the slide are shown on an enlarged scale in Figs. 228 and 229. The end of the tiller is made rectangular in section and perfectly true and parallel. A carriage or frame, fitted with a brass swivelling block, runs on guides carried from side to side of the ship, at right angles to the longitudinal centre-line. This swivelling block fits over the rectangular end of the tiller, and is arranged to allow a certain amount of rotatory motion in a horizontal plane. The frame or carriage is worked to the right or left as required, by chains or ropes from the steering gear. As the rudder

Fig. 228.



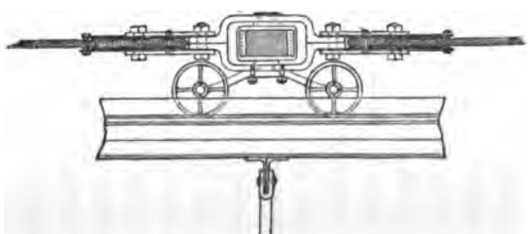


Fig. 229.

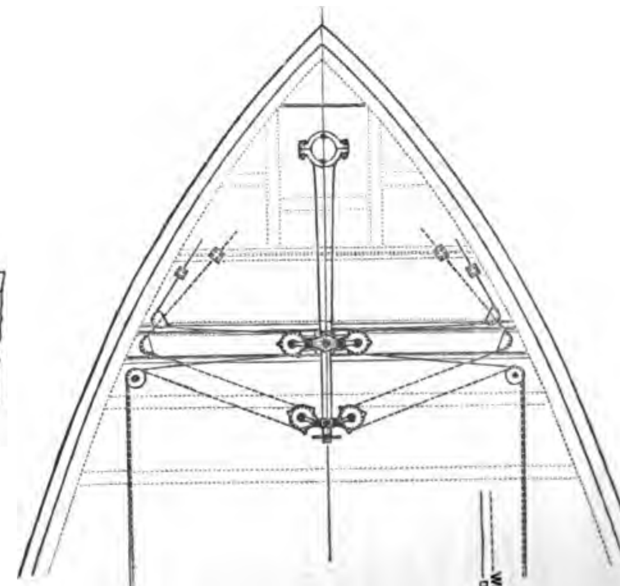


Fig. 227.

is turned, the end of the tiller slides in the block, and thus increases its effective length; the perpendicular distance between the centre of the rudder-head and the direction of the force remaining constant, so that the turning moment is uniform for all angles of the tiller.

Capstan engine.—When the capstans of ships are fitted to be worked by steam, the power is usually applied through the medium of a worm wheel, keyed on the lower end of the spindle, which is driven by a worm or endless screw worked by the capstan engine. In some cases a large spur or crown wheel is fixed on the capstan spindle and driven by a toothed pinion on the capstan engine shafting. The capstan spindle is carried through the deck, and its weight is supported by a suitable step or bearing below.

In some instances the principal pumping engine of the ship has been utilised for working the capstan, by fitting shafting and gearing to work the worm or pinion that drives the capstan. In these cases, disconnecting gear should be fitted to enable the capstan to be released when the engines are required for pumping purposes only. More generally, however, independent engines are fitted for driving the capstans. These are of various types in accordance with the designs of different makers, but it is unnecessary to enter into detailed descriptions. All of them must be fitted with reversing gears, so that the capstan may be revolved in either direction as required. In some cases this is effected by reversing the motion of the engine itself; whilst in others the motion of the capstan is reversed by an arrangement of friction clutches, the engine working in one direction only. This plan, however, is not so suitable and safe as the fitting of reversing gear to the engine, and requires to be used with caution, as during the operation of reversing there is an instant at which the capstan is free and the

load, if any, is liable to *take charge* before the clutches get sufficient grip to give the reverse motion. When the capstan is being worked by steam power the safety pawls on the capstan-head should be kept lifted to prevent accident, the gearing of the engine itself being sufficient to hold the capstan in any position.

The cylinders of all auxiliary engines should be made large enough to enable the engines to be efficiently worked with reduced pressures of steam. In the case of the capstan engine, it is necessary to fit *reducing valves* in the steam-pipes to prevent the possibility of excessive strains being brought on the chain cables when high-pressure steam is being used. These reducing valves should be so designed and arranged that the engine may not be able to produce a pull of more than two-thirds of the proof strain of the cable whatever the pressure of steam in the boilers may be.

Turret-turning engines and gear.—A general arrangement of the steam-engines and gear fitted for the purpose of revolving the gun turrets is shown in Fig. 230. In the plan shown in the drawing, a pinion wheel, A, slightly bevelled, which is driven by the turret engine through the medium of suitable shafting and gearing, works into a toothed rack fitted underneath the turret close to its outer edge. In this arrangement the pinion and rack form, in point of fact, a pair of bevel-wheels, the diameter of the one on the turret being very large as compared with that of the driving wheel. In some instances racks with vertical teeth are fixed outside the turret, so as to form large spur wheels into which ordinary toothed pinions gear. A pair of racks and pinions is usually fitted in such cases to lessen the liability to accident. All the turret-turning gear should be made very strong and substantial, to withstand the shocks that may be brought on it by the sudden reversing of the engines when the heavy turret is revolving at full

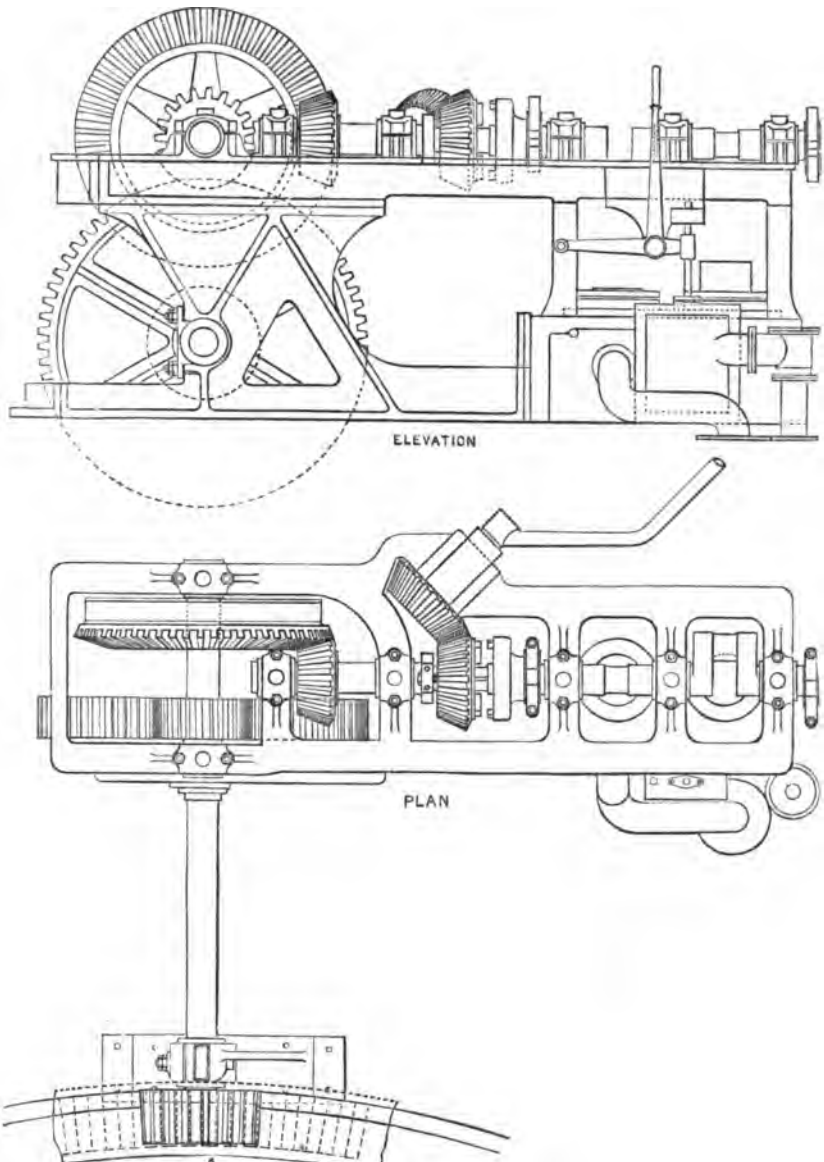


Fig. 230.

speed. The racks on the turrets are made of strong gun-metal, with shrouded teeth to increase their strength, and the driving pinions are usually made of wrought iron or steel. The arrangement of the gearing through which the power of the engine is transmitted to the turret is clearly shown in the diagrams, and needs no detailed description.

The reversing valve of the turret engine is usually a balanced differential valve, similar to that described at page 267; and in addition to the starting lever at the engine, gear should be fitted to enable the reversing valve to be worked from inside the turret by the officer setting the gun. The lever at the engine is used when the guns are set by director. An index or pointer is fixed to a bracket on the engine framing, or on the deck near the turret engine, and the lower edge of the turret is graduated in degrees in such a manner that the operator at the engine is enabled to set the guns at any desired angle with the line of keel, by observing the angles marked on the turret as it revolves.

The turret revolves round a hollow central pivot bearing, the weight of the turret being carried on rollers, near its outer circumference, running on an accurately turned roller path. The roller path is generally made of wrought iron or steel. In some instances manganese bronze has been used for this purpose. The gearing for working the turning engine from inside the turret is carried to the engine through the central pivot. To prevent water passing down between the turret and the deck, leather flaps are attached to the circumference of the turret above the deck, which slide over the glacis plates fitted round the turrets. These surfaces should be frequently lubricated to prevent the rapid wearing away of the leather.

Hydraulic turning gear for turrets.—In the larger turret ships, hydraulic machinery has been employed for

working the turrets, &c. In this arrangement two duplicate steam pumping engines are fitted to produce the water pressure, which is conveyed by suitable piping to the turret-turning engines, ammunition and shot lifts, loading and elevating gear, &c. A uniform water pressure of about 800 pounds per square inch is maintained in the

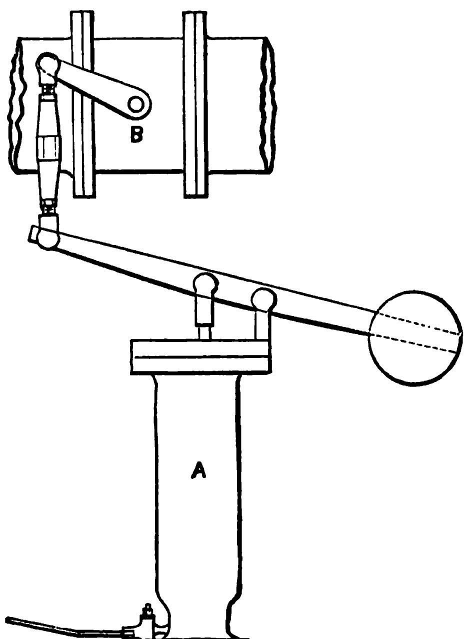


FIG. 231.

pipes by the agency of a piston in a cylinder, the pressure of the water being admitted to the bottom of this cylinder by a small pipe led from the main pressure pipe, and this pressure is resisted by the action of a spring of suitable strength on the top of the piston. A rod from this piston is connected by levers to a throttle-valve in the steam-pipe

leading to the pumping engine, so that the pumps are started or stopped as may be required. A sketch of this arrangement is shown in Fig. 231, A being the water cylinder and B the throttle-valve in the steam-pipe. A spring-loaded relief or escape-valve is fitted on the main pressure pipe, which allows the water to escape to the supply or suction tank for the pumps, when the pressure exceeds 800 pounds per square inch. The hydraulic turret-turning engines are each fitted with three oscillating water-pressure cylinders, the pistons of which act on cranks at equal angles to each other, and the motion is transmitted to the turrets by means of shafts and toothed gearing, as in the steam turning gear.

In addition to working the turret-turning engines the water pressure is generally utilised for working a number of hydraulic lifts and machines for loading, training, and working the guns, &c. As before pointed out, this system might, in many cases, be extended with advantage to much of the other auxiliary machinery of ships, so as to promote uniformity and simplicity in design and arrangement.

Hand turning gear is generally fitted in addition to the steam or hydraulic gear, to provide for the event of accident to the turning engines, and to enable the turrets to be worked by manual labour for preservation and for gunnery practice when steam is not raised in the boilers.

Air-compressing machinery for torpedoes.—Machinery for compressing air for the purpose of charging and launching Whitehead torpedoes is now fitted in nearly all war-ships. We purpose giving a general description only, as it will be unnecessary in this book to enter into details. The types of engines and pumps used for this purpose are varied in design and arrangement, but they all consist essentially of two or more pumps with successively de-

creasing diameters. Air is drawn from the atmosphere into the largest pump and is compressed successively through the intermediate pumps, till it finally issues from the smallest pump and passes to the air reservoir, through a coil of small copper pipe surrounded with cold water to reduce the temperature of the compressed air. Cold water is kept circulating round the several pump-barrels to carry off the heat produced by the compression of the air; and a small stream of water and oil is admitted to the largest pump-barrel, with the air, to lubricate the leather packings of the pump-plungers, and to assist in the reduction of the temperature. A *separator column* is fitted close to the pumps to separate this water and oil from the air, and prevent its entering the reservoir; a drain cock, fitted on the bottom of the column, being opened at intervals to get rid of the water, &c. Arrangements are now being fitted for discharging torpedoes by the explosion of gunpowder behind them.

Air reservoir.—The principal *air reservoir* generally consists of a group of 50 steel tubes, connected together at their ends by small copper pipes and screwed unions. The tubes are about 3 inches external diameter, $\frac{3}{16}$ths-inch thick, and 6 feet long; the total capacity of the reservoir being about 11 cubic feet. These reservoirs and their connections are all tested by hydraulic pressure to 2,250 pounds per square inch, the maximum working air pressure being 1,500 pounds per square inch. The engines and pumps in large ships should be of sufficient capacity and power to compress 30 cubic feet of air to a pressure of 1,500 pounds per square inch in an hour. From the main reservoir, seamless copper pipes, with a bore of about $\frac{3}{8}$ths-inch, are led to the *charging columns*, fixed in various parts of the ship, from which branches are led for charging the torpedoes, firing carriages, &c., with compressed air.

It is obvious that in all torpedo gear and fittings the greatest reliability and accuracy are necessary, as such very high pressures and sudden impulses have to be dealt with. Every precaution must be taken to prevent leakage, and too great an amount of care cannot be exercised in the fitting of every detail of the gear.

Boat hoisting engines.—In most large ships engines arranged to drive suitable winch barrels are now usually fitted for the purpose of readily placing the torpedo and other boats into the water, or of lifting them out of the water and landing them in their proper positions on board, in fixed crutches. Two winches are necessary, one for lifting the boats out of the water, and the other for topping the derrick and bringing the boats inboard. The derrick is hinged on a swivel or ball joint, supported by a bracket secured to the mast or signalling pole. There are many different arrangements in the details of the engines and winches fitted for this purpose: but it is desirable that in every case the engine and winch barrel used for topping the derrick should be independent of that employed for lifting.

Ventilating engines.—In all turret ships, and in many other vessels, it is necessary to fit blowing engines for the purpose of providing a supply of fresh air for the respiration of the crew as well as for the ventilation of the engine and boiler rooms. The engines are usually arranged to drive rotatory fans or blowers, which draw air from suitable hatchways, or from special ventilating shafts carried up to the flying deck, and distribute it through the ship by means of ventilating trunks or pipes with suitable openings in the several compartments. The outlets from these ventilating pipes are generally fitted with light gridiron valves, so that only such may be opened at any time as may be required. In modern war-ships, with closed stokeholds, blowing-fans and engines are also fitted

for forcing the draught in the boilers as described at p. 65.

Edmond's ventilating system.—In this arrangement, air trunks or channels are carried along the sides of the ship or underneath the beams, with grated openings covered by suitable sliding doors, in the several compartments. These air trunks are connected at various parts of the ship with upcast ventilating shafts, into which jets of steam are introduced to quicken the upward current of air through them. When the steam jets are acting, the foul air is drawn from the several compartments into the air trunks and expelled through the ventilating shafts; its place being supplied by fresh air from the hatchways, or from special downcast shafts if necessary.

Ash-hoisting engines.—Small engines to drive winch barrels are fitted in most steamships for the purpose of lifting the filled ash buckets, &c., from the stokeholds, to enable the ashes to be thrown overboard. Some of these engines are fitted with reversing gears, so that the buckets are both raised and lowered by the working of the engines: but, in the majority of cases friction or clutch gear is fitted to allow the barrel to be disconnected from the engines so that the bucket may descend by its own weight, the rate of descent being regulated by a break. Railways are fitted on the upper deck, from the top of the ash-tube to the shoots at the ship's sides, to facilitate the discharge of the ashes, &c.

Feed and bilge pumping engines.—The pumping engines fitted for feed and for bilge or fire purposes are usually similar to each other in design and construction, and two or more of each are fitted in all ships. A sketch of pumping engines of this description, with two cylinders and double-acting pumps, is shown in Fig. 232. The feed donkeys should have no connection with the bilge, and two are generally fitted in each stokehold, one

to pump from the feed-tank and condenser only and the other from the feed-tank and the sea. With this arrangement, feed-pumps worked direct by the main engines are not necessary. Double cylinders are desirable for these engines to facilitate starting; and the slide-valves are

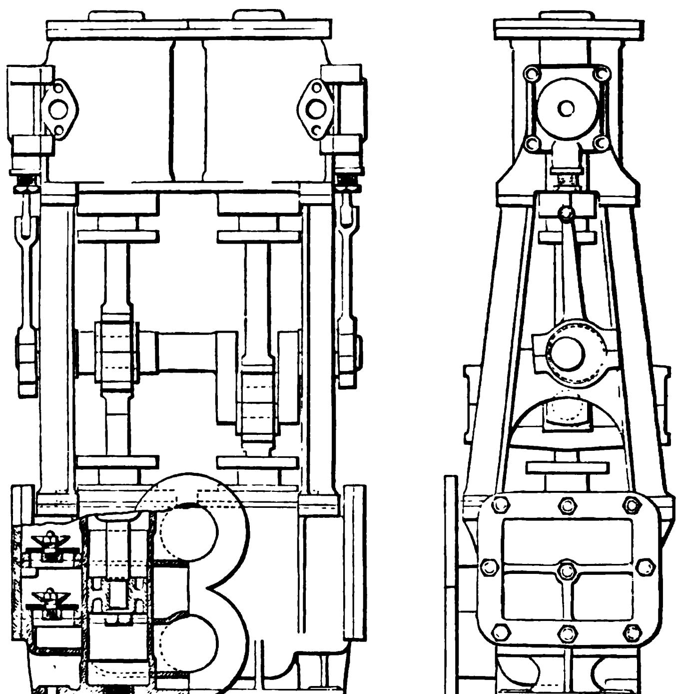


FIG. 232.

made with very little lap to insure the engines starting readily in any position of the cranks; economy in the use of the steam being in these cases a minor consideration.

Silent blow-off to condensers.—In the event of a naval engagement it is probable that it would be necessary to

make sudden variations in the speed of the ship or even to stop altogether for a time, with the engines ready to start at full speed when ordered, so that the fires would require to be kept burning briskly. To prevent, on such occasions, a rush of steam through the waste steam-pipes, which would cause much noise and inconvenience on deck, and prevent the orders given from being heard, direct communication is made between the main steam-pipes in the engine-rooms and the engine condensers, to enable the surplus steam to be blown direct into the condensers. This is usually known as the '*silent blow-off*.' In the pipe connecting the steam-pipe to the condenser a valve is fitted which is worked from the starting platform, and a suction-pipe from one of the pumping engines is led to the bottom of the condenser to enable the condensed steam to be pumped into the boilers. Steam yachts have sometimes been fitted with valves to allow steam to be blown direct overboard when the engines are standing with full steam up ready to start; but blowing the steam into the condensers makes less noise on deck and prevents loss of fresh water, which is very important with high-pressure boilers.

These fittings also enable the main engine condensers to be readily utilised for distilling fresh water for drinking and culinary purposes, in cases of emergency, such as those that have frequently arisen in many warlike expeditions.

Auxiliary condenser.—In the earlier ships the exhaust steam from the auxiliary engines was led into the main engine condensers, as well as into the waste steam-pipes, in order to prevent loss of fresh water. The number of auxiliary engines in modern war-ships is so great that the unavoidable leakage from the several engines would seriously affect the vacuum in the main engine condensers, and whenever the main engines were not at work the auxiliary engines would have to exhaust into the atmo-

sphere, and thus cause considerable waste of fresh water. To remedy these defects separate auxiliary surface-condensers have been fitted in recent ships to receive the exhaust steam from all the auxiliary engines of the ship. These auxiliary condensers are fitted with independent circulating pumps, and also with air-pumps that deliver into the feed-tanks, so that the condensed steam may be returned to the boilers, by the feed-pumps.

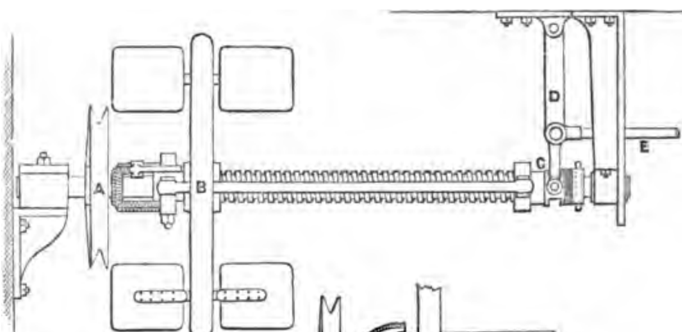
Governors.—The object of the governor is to maintain uniformity of motion of the engines when the resistance experienced is varied from any cause. In a land engine this variation usually results from the number of machines that may be put in or out of operation at any time. In a marine engine, considerable variation of resistance may ensue from the pitching motion of the ship, which alters the immersion of the propeller, and, in the absence of any regulating or governing apparatus, would cause what is technically called *racing* of the engines, the revolutions increasing when the stern is raised, and diminishing when it is depressed.

The governor for a land engine usually consists of a pair of heavy balls, suspended on arms jointed at or near the axis of a vertical rod, which is rotated by the engine. As the revolutions of the engines are increased, the balls fly outward under the action of centrifugal force and work a system of rods and levers, connected with the throttle or expansion valve, so as to reduce the amount of steam admitted to the cylinders. If the revolutions be decreased from any cause, the balls will drop and the throttle or expansion valve, as the case may be, is opened wider. In this system, therefore, the balancing forces are due to the centrifugal action and the force of gravitation on the revolving balls. This, however, would not be suitable for marine purposes, as the action of gravitation on the heavy balls would be affected by the motion of the ship, so that

the balance of forces, on which the operation of the machine depends, would become irregular. Consequently, in marine engine governors the action of a spring is substituted for the effect of gravity, in order to enable the force acting on the regulating valve to vary uniformly with the revolutions of the engines. Many different arrangements have been adopted for this purpose, but we must content ourselves with the description of two or three only, to illustrate the general principle of their action.

Silver's governor.—This was one of the earliest plans adopted for marine engines, and proved to be very efficient.

FIG. 233.



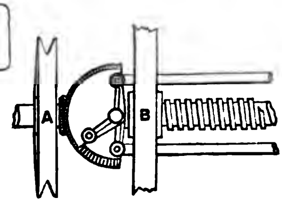


FIG. 233*a*.

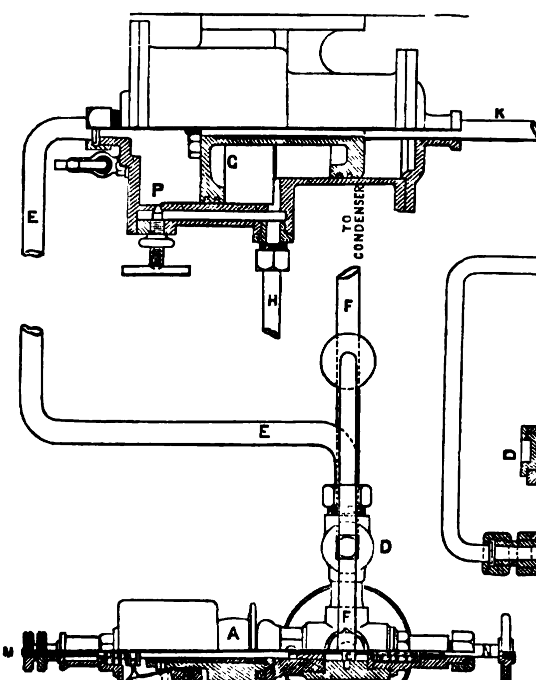
A sketch of this governor is shown in Fig. 233. It is driven from the engine or propeller shafting by means of a hide-rope belt passing round the pulley A. The heavy fly-wheel B, fitted with vanes, is loose on the shaft, and is worked by the action of a bevel-wheel on toothed segments attached to the fly-wheel. A plan of these segments and fittings is shown in Fig. 233*a*. Rods are

carried from these segments to a loose collar C, which is connected by means of the levers D and E to a throttle-valve in the main steam-pipe. A steel spiral spring is fitted between the collar C and the boss of the fly-wheel B. When the engine is working uniformly, its motion is transmitted to the fly-wheel through the bevel-wheel and segments, and a certain compression is thereby produced in the spring. If the motion of the engine be suddenly accelerated or retarded from any cause, the inertia of the fly-wheel will tend to keep it still moving uniformly, and the change produced by the engine in the speed of the pulley A will alter the relative positions of the segments, and thus act on the throttle-valve through the system of rods and levers, so as to open or close it as may be required.

Governors of this description, which act *directly* on the throttle-valve, require to be made very large and heavy, as the motion produced on the valve is due entirely to the work accumulated in the fly-wheel. They therefore absorb a considerable amount of power in working, and their action is not sufficiently rapid for modern compound engines working with high-pressure steam. To decrease the weight and increase the sensibility of governors for marine engines, the more recent instruments have been designed to cause the revolving part of the apparatus to actuate a small valve only, for the purpose of admitting steam to a cylinder, in which a piston works, connected by rods and levers to the regulating or throttle-valve. In these arrangements steam pressure is utilised for working the regulating valve, so that the governing apparatus, which only has to work a small balanced valve, may be made very light, and the rapidity of action of the gear is much increased.

Westinghouse governor.—The governing apparatus in this arrangement is shown in the lower part of Fig. 234. It is driven from the main shaft by means of a rope or

leather belt passing over the pulley A. If the speed of the engines should increase, the weights, B. B, will fly out and cause the small piston C to uncover the port and open a communication, through the pipe F, between the condenser and the top of a small valve at D, which is shown in section in Fig. 234*a*. In consequence of the vacuum in



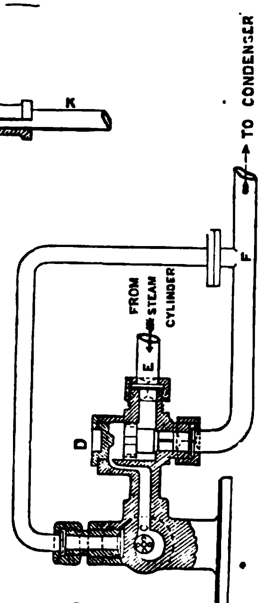


FIG. 234a.

FIG. 234.

the condenser, this valve D lifts and places the back of the steam piston G in direct communication with the condensers through a second branch leading into the pipe F, which had been previously closed by the valve D. The throttle-valve is actuated through rods and levers, by means of a compound piston G, in a steam cylinder, the

larger end of which is connected to the governing part of the apparatus through the pipe E, and consequently to the condenser when the valve D is opened so as to make communication between the pipes E and F. Steam is admitted from the boiler, through the pipe H, to the central part of the piston G, and also to a limited extent through the screw-valve P, to the back of the piston, so as to keep it in equilibrium. When, therefore, from increase of speed of the engines, the weights fly out and cause communication to be opened between the back of the piston G and the condenser, the steam pressure forces the piston to the left and cuts off the admission of steam to the engines, by closing the throttle-valve, to which it is connected by means of the rod K. When the engines resume their normal speed, the action of the springs in the governing part of the gear will bring the weights B B to their original positions, so that the piston C will again cover the port and cut off the communication with the condenser. Steam will then enter the steam cylinder and press the piston to the right; thereby opening the throttle-valve and again admitting steam to the engines. The adjustment of this governor to suit any required speed of the engines, is effected by regulating the compression of the springs by means of the screws M and N: the former being set before the engines are started, and the latter being used to correct any deviation from the desired speed when the engines are at work, to insure accuracy.

Durham's velometer.—Sketches of this form of marine engine governor, which is fitted to a large number of steamships and has proved to be very efficient, are shown in Figs. 235 and 236. In this arrangement, A is a closed cylinder, kept full of water, which revolves around a fan with 6 or 8 vanes which is fixed inside it; the water cylinder forming a fly-wheel and the fan a constant break on it. The machine is driven by the pulley B, and the rotation is transmitted to

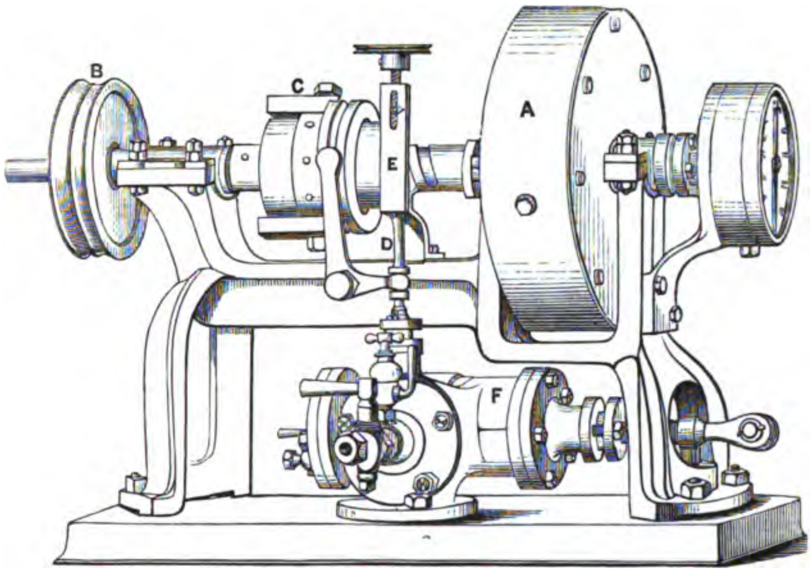


Fig. 235.

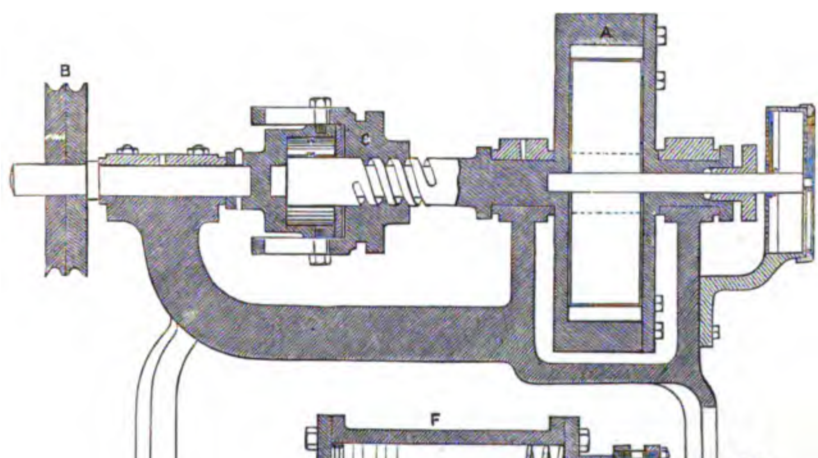


Fig. 236.

the water cylinder through a coupling nut C, which works over a screw thread cut on the shaft carrying the cylinder A. The coupling C is connected by a bell-crank lever to a rod D, the upper end of which is acted on by a spiral spring in the barrel E. The position of the coupling nut C, on the screw thread, is regulated by the tension of the spring in this barrel, which varies with the amount of resistance produced by the fan on the water cylinder as the speed increases or decreases. F is a steam cylinder, carried on a pivot to enable it to be turned round to the direction required to allow the piston-rod to be connected direct to the governor throttle-valve without intermediate gear. At the lower end of the rod D, a small slide-valve without any lap is attached, so that the raising or lowering of the rod, by the action of the governor, causes steam to be admitted to either side of the piston as required; and the throttle-valve is opened wider, or closed, as may be necessary to produce regular working of the engines. The velometer is regulated, for any required speed of the engines, by setting the adjusting screw at the top of the spiral spring box E. In the more recent instruments of this kind, a dial or *speed indicator* has been attached for showing at a glance the number of revolutions per minute at which the engines are running. In this case, the fan is held in position by a second spring in the speed indicator gauge, so that the force of water thrown by the revolutions of the cylinder itself is measured and indicated by a pointer on the dial.

Dunlop's governor.—The governors previously described are brought into action by the increase or decrease in speed of the engines. Dunlop's patent pneumatic governor is designed to anticipate and prevent this variation of speed, and is therefore more perfect in principle for general work at sea. It consists of an air chamber at the after part of the ship, which is in connection with the governing apparatus by means of a small pipe. As the propeller

rises and falls in the water, the air pressure in the chamber and pipe varies, and moves a small valve which admits steam to one or the other side, as required, of a steam piston which actuates the regulating valve for the engines. The admission of steam may thus be proportioned to the immersion of the propeller, and the racing of the engines in rough weather prevented. The reports of the action of this kind of governor in ocean steamers have been very satisfactory.

Distilling apparatus.—The water used on board ship for drinking, washing, and culinary purposes, &c., is produced by the distillation of steam. In addition to its convenience, there is little doubt that this practice has added greatly to the health of the Navy, as the water thus obtained is perfectly pure, which it is almost impossible to insure being the case in many ports at different parts of the world.

Normandy's condenser.—The type of distilling condenser most generally employed in the Royal Navy is that patented by Dr. Normandy, a sketch of which is given in Fig. 237. Steam from the boiler is admitted to the condenser through the pipe A, and passes through the two series of tubes shown, the upper set being the condensing and the lower the cooling tubes. The casing is kept filled with cold sea-water, which enters at the orifice B, at the bottom, and flows out through the upper orifice C. An air-pipe D is fitted to allow the air, evolved from the condensing water in the casing by the heat, to pass into the condenser tubes with the steam so that the water produced may be properly aërated. This pipe should be carried up to at least 6 feet above the water level, to prevent the possibility of any sea-water passing into the condenser. The pipe E leading from the chamber between the condensing and cooling tubes, is fitted to allow any excess of air to pass to the atmosphere. It also serves as an in-

dicator of the maximum amount of steam that should be permitted to enter the condenser, for if more be admitted than can be condensed by the upper set of tubes, steam will rise through the pipe and come out through the orifices at

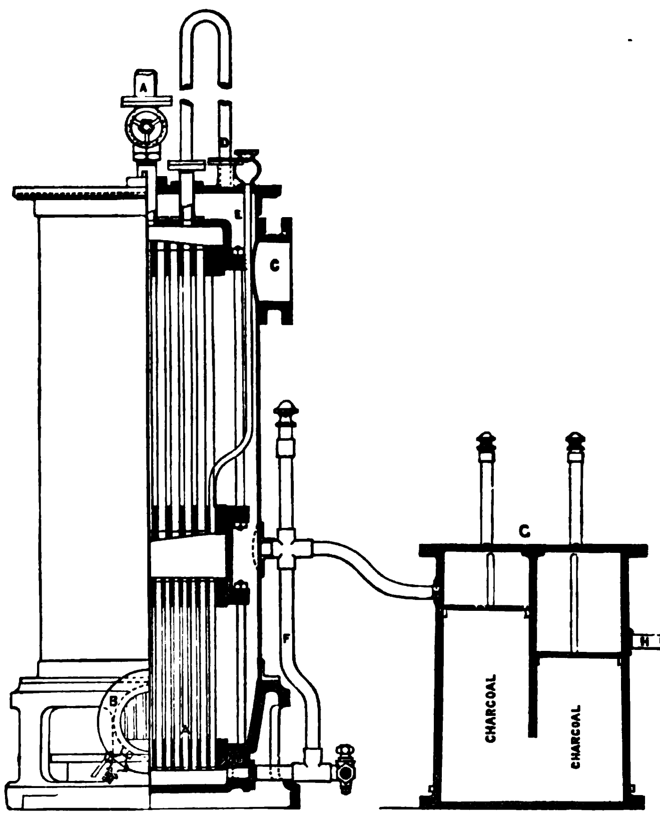


Fig. 237.

the top. In this case the steam inlet valve should be partially closed until steam ceases to issue. The condensed water rises from the lower chamber through the pipe F and passes to the filter G, in the centre of which a diaphragm

plate is fitted to cause the water to pass through a mass of granulated animal charcoal, down to the bottom of the filter and up on the other side to the outlet pipe H, which leads to the tanks. The top of the pipe E is covered with a perforated knob, and air-pipes are fitted on top of the filter to allow any excess of air to escape. These condensers are very efficient, the water produced being quite cold, aërated, and fit for immediate use, on being drawn direct from the filter.

In a large number of merchant steamers, and in some ships of the Royal Navy, other kinds of distillers have been fitted. Kirkaldy's compactum condenser and Hocking's condenser which, in addition to Normandy's, are the most generally employed, have proved very efficient and effect a considerable saving in space.

Distilling condensers are generally fitted below the water-line; so that when the inlet and outlet valves are open the casing is always quite full of sea-water, and a natural circulation is kept up when the condensers are at work, by the heated water passing out at the upper orifice and the cold water coming in at the bottom, to supply its place. To increase the flow of the water through the condensers, a suction-pipe from one of the auxiliary pumping engines in the ship is usually connected to the outlet orifice of the condenser, to enable the water to be pumped through, and this, if properly arranged, increases both the rapidity and the economy of the distillation. In recent ships, the large inlet and outlet pipes and valves to the ship's side required for natural circulation of the water, have been dispensed with, and the circulation produced entirely by the pumping-engine. The distilled water flows into receiving tanks close to the condensers, and from these tanks it is pumped by means of a small steam donkey-pump to the water tanks in the hold of the ship.

Steam fog-whistle.— All steam-ships are fitted with

steam whistles, which may be used for signalling purposes and also for indicating the position of the ship in case of fog, &c. A sketch of this instrument is shown in Fig. 238. The sound is produced by the vibration caused by the action of the film of steam issuing from the narrow annular orifice, against the thin edge of the bell of the whistle. The bell is screwed on the upper part of the spindle, to enable its distance from the steam orifice to be readily adjusted to suit the pressure of steam used. The steam pressure acts on the top of the valve and tends to keep it closed. A spiral spring is also fitted to assist this action. The valve is opened by a lever, from the ends of which a string or wire is usually led to some convenient place for working the valve and blowing the whistle.

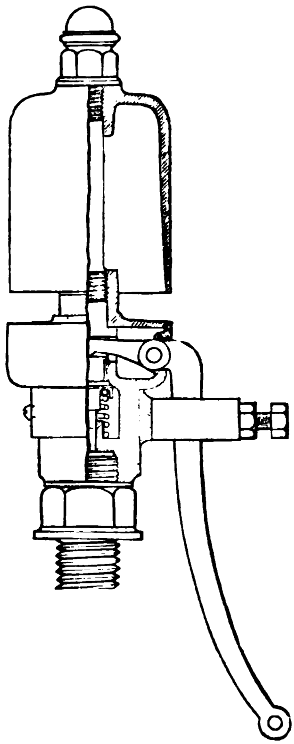


FIG. 238.

Holmes' patent siren is another instrument by means of which steam is used for signalling purposes at sea. The steam passes through a number of narrow slits on the surfaces of two small horizontal cylinders one of which revolves within the other, and issues to the atmosphere through a bell- or trumpet-mouthed orifice made of hammered sheet copper or brass to increase the volume of sound. It is much louder than the ordinary fog-whistle, and, in addition to signalling, this instrument

may be used for discovering the position of other vessels, rocks, land, &c., when hidden by fog. This is done by turning the bell of the *siren* in different directions successively, and sounding sudden sharp signals. If any high object should be within a radius of about two miles the sound would be echoed back when the bell of the instrument was turned in the direction of the object; and its distance from the ship could be roughly estimated by the interval between the sound and the echo; one second representing approximately one-tenth of a mile.

Telegraphs.—The necessary orders from the deck to the engine-rooms, and to the principal auxiliary engines of the ship, are usually transmitted, by means of small shafting and gearing, to index hands or pointers which work over dials suitably engraved. The construction of the dial-boxes for all the telegraphs is the same, but the orders engraved on the respective dials are necessarily different. A loud gong is fitted in the apparatus, the hammer for which is worked by a sprocket wheel on the index spindle, to call attention to the moving of the telegraph. Reply gongs are fitted to indicate to the officer on deck that the order has been received and acted upon.

Electric telegraphs.—In some ships electric telegraphs have been fitted for transmitting orders, but hitherto the mechanical details of this system have not been sufficiently perfect to cause its general adoption. It would, however, be a very great advantage if electric currents could be utilised for this purpose, especially in ships in which communication has to be made between several compartments at considerable distances from each other; and it is not improbable that before long the difficulties that have hitherto been experienced in the arrangements of the details of the gear, will be successfully overcome, and that electric telegraphy will become practicable and reliable for transmitting all the necessary

orders on board ships. In the present tentative stage of the question it is undesirable that space should be occupied with descriptions of the plans that have been under trial.

Voice-pipes.—To facilitate communication, suitable voice-pipes, about one inch in diameter, with whistles and mouth-pieces at their ends, are also fitted from the deck to the main and auxiliary engine-rooms, and between such other parts of the ship as may be considered necessary.

Tell-tale apparatus.—Standards and dials, with revolving pointers or arms, driven by the main engines through shafting and gearing, are fitted in convenient positions on or near the bridges, and in the conning towers, to enable the officers on deck to satisfy themselves as to the speed and direction of motion of the engines at any time.

Herreshoff boiler or steam generator.—The Herreshoff system of machinery, which has, as yet, been applied principally to torpedo-boats, pinnaces, and other small vessels, may almost be considered, in many respects, as a new departure in marine engineering. It is quite original in most of its arrangements, and possesses many features of interest and importance.

The principal element in this system is the boiler, which consists of two spiral coils of iron piping connected together at the top, and arranged as shown in Fig. 239. The outer one is of uniform diameter throughout, and is called the *feed-water coil*; the inner coil is the *steam generator*, and is made of pipes welded together, and gradually increasing in diameter towards the lower end. The two coils are carried on a dwarf wall of firebrick which forms the boundary of the furnace. In the inner coil, or steam generator, the convolutions are placed close together at the top, but at the sides narrow spaces are left between them; whilst in the outer or feed-water coil, the convolutions are

close together at the sides and separated at the top. The object of this is to cause the heated gases to pass out through the divisions of the inner coil to the space between the coils, and thence to the funnel through the openings at the top of the feed-water coil, so that the steam-generating coil is wholly surrounded by the hot gases, and the heat as

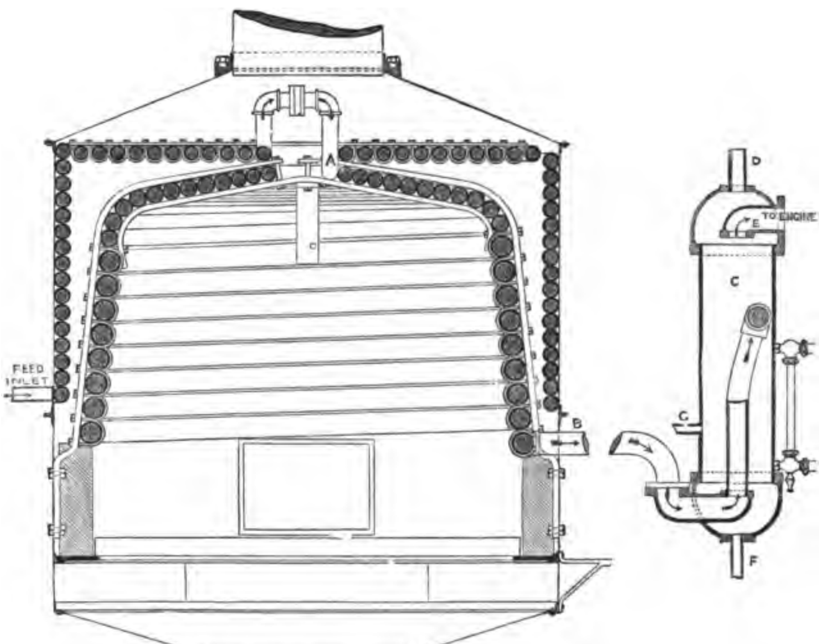


FIG. 239.

fully utilised as possible. The space above the fires for the combustion of the gases is very large, which tends to promote economy of fuel. A casing of thin sheet iron or steel is fitted around and above the feed-water coil, the upper part being clothed with silicate cotton or other non-conductor, to prevent radiation.

The peculiar feature of this boiler is, that the feed-

water is admitted at the top of the steam generator, and being vaporised during its descent passes off as steam from the bottom, which is quite the reverse of the arrangement in ordinary boilers. The feed-water, which is heated to some extent by passing through the outer coil, enters the steam generator at the place marked A at its extreme upper end, Fig. 239, whence it flows, partly by gravity but mainly by the action of the *circulating pump* (which is peculiar to this system), down to the lowest convolution, being, to a greater or less extent, converted into steam during its descent. If the supply of feed-water relatively to the heat produced by the furnace be such that the water is evaporated in the upper part of the coil, it will be superheated in the lower part and the cylinders and the packings in the glands are liable to injury from the hot dry steam. A surplus quantity of feed-water, above that required to be evaporated, is therefore caused to circulate through the coil, so that the steam, on leaving the boiler, is super-saturated, or mixed with a proportion of water. This mixed steam and water is carried through the pipe B into the separator C (which in this system is an indispensable adjunct to the boiler), where the water separates from the steam and falls to the bottom, whilst the steam rises and passes out at the top into the steam-pipe leading to the engines. The separator serves as a steam-chest to the boiler, keeping the pressure practically uniform during the intermittent passage of steam to the cylinders of the engines. From the top of the separator an iron pipe D is led to the safety-valves, and another, E, to the main stop-valve near the engines. A water-gauge glass is placed at the lower end of the separator, and serves the same purpose as the water-gauges on ordinary boilers. In the present case, however, it indicates the amount of surplus feed-water pumped through the steam generator above that required to be evaporated, and enables this to

be regulated to the quantity necessary to insure the steam being delivered in the most suitable condition as to saturation. Two blow-out pipes are fitted to the separator, one, F, at the bottom to allow the sediment, &c., to be blown overboard, and the other, G, called the surface blow-out, is fitted on the side, at the height at which the water should be kept. A suction pipe is led from this water level to the circulating pump, to enable the surplus feed-water to be again pumped through the steam-generating coil. An additional valve and branch pipe is fitted to connect this suction pipe direct to the condenser when necessary to get rid of an excess of circulating water.

The engines in the Herreshoff boats are usually surface-condensing; the condenser consisting of two copper pipes running along the bottom of the boat, one on each side of the keel. These pipes continuously decrease in diameter from their junction with the eduction pipe to the opposite ends, from which the condensed water is drawn by the air-pump and pumped into the hot-well. From the hot-well the feed-pump draws its supply and delivers the water into a pipe which, together with the blow-out pipe from the separator, leads into the suction of the circulating pump. By this arrangement the circulating pump works almost in equilibrium, as the pressure in the suction pipe is necessarily equal to the steam pressure in the separator, to which it is connected by the surface blow-out pipe, and the only difference in pressure between the suction and delivery pipes of the circulating pump is that required to force the water through the feed-water and steam-generating coils. A fresh-water supply tank is fitted, from which a suction is led to the air-pump to make up for any waste resulting from leakage of glands, blowing of steam whistle, &c. An overflow pipe is fitted to the hot-well, so that if the water should accumulate, it may escape to the supply tank.

If an excess of water is being circulated through the coils it will be shown by the rising of the level in the separator gauge glass; a part of it is then caused to pass direct from the separator to the condenser by means of the additional valve and pipe before mentioned, instead of all going to the suction of the circulating pump. If insufficient water be supplied, so that the steam has little or no water mixed with it on entering the separator, the quantity of water discharged by the feed-pump to the circulating pump suction must be increased.

There are other minor peculiarities in the Herreshoff system, as applied to boats, which we have not space to enter into; but it will be seen by the brief description given that all the arrangements and operations in this system differ very considerably from ordinary practice. The peculiar feature of the system is the coil-boiler, which possesses the advantages of safety under all conditions, simplicity of construction, rapidity of generation of steam, and a total absence of priming. The safety of these boilers may be considered almost absolute. Explosion, in the ordinary sense of the word, is impossible, for the quantity of water carried is so small that if it were instantly liberated no damage could ensue, and the splitting of any part of the coil would only result in the harmless blowing of steam up the funnel. The separator is the only part of the apparatus of large diameter, and this is made with a large factor of safety and is entirely removed from the action of the fire. The damage following ordinary boiler explosions is mainly due to the sudden evaporation of the large body of water at high temperature contained in the boiler, which condition is entirely absent from the Herreshoff system. This coil-boiler contains the smallest quantity of water and is the lightest yet constructed, for the same power, which is important, not only for torpedo-boats, but also for all small steam-boats carried

by war-ships, as the weight saved in the machinery could be utilised for additional coal stowage to enable the boats to be self-supporting for longer periods. With these generators steam can be raised with great rapidity, only about six minutes being required from the time of lighting fires to raise steam of sufficient pressure to start either the auxiliary pump, or the main engines, to continue the circulation through the coils.

CHAPTER XXVII.

CARE AND MANAGEMENT OF THE MACHINERY.

We purpose in this chapter to embody in a compact form a few general particulars relative to the care and management of the machinery and boilers of ships, which are to a considerable extent based upon the instructions contained in the 'Steam Manual' now in use in the Royal Navy.

Filling the boilers.—When preparing to raise steam, the boilers are usually, in the first instance, filled with water to some distance above the ordinary working level, to allow for the evaporation that ensues during the warming of the engines, and for the blowing out of some of the water to promote circulation. If fresh water be used, the boilers are filled by means of a hose through one of the upper manholes; but when sea-water only is available, the boilers may be run up by opening the bottom blow-out and sea-cocks, as the whole of the boilers are generally below the water-line. During this process the safety-valves, gauge-cocks, &c., should be opened to allow the air to escape from the boiler as the water flows in. It is generally advantageous to fill the boilers with fresh water, if it can be conveniently obtained, and every care should be taken to avoid the necessity of running water into the boilers from shallow depths, or for using impure water of any kind.

Laying fires.—The fire-grates of the boilers that are

intended to be used are covered with a layer of coal, and firewood, shavings, oily waste, etc., are heaped up at the mouths of the furnaces over the dead-plates. As the wood burns up coal is added to it, and small fires are thus made at the mouths of the furnaces. During this process the ashpit doors are kept closed and the furnace doors open, so that the heat and flame from the fires at the entrances are carried over the coal laid in the furnaces, and tend to ignite it. When a sufficient quantity of coal is alight, the fires at the furnace mouths are pushed back from the dead-plates into the body of the furnaces, and spread over the partially-ignited coals on the bars; the furnace doors are then closed and the ashpit doors opened as required, to allow the air for the combustion of the coal to pass through the fires from underneath in the ordinary manner.

Raising steam.—During the earlier part of the process of raising steam the safety-valves should be kept partly open, to allow the air to escape from the boilers into the atmosphere. When steam pressure begins to show in the gauge the safety-valves should be closed; but they should be moved occasionally during the time the steam pressure is being increased, to insure their remaining in good working order. The fires should be kept low, so that the steam may be raised very gradually, to allow the different parts of the boilers to expand, from the increase in the temperature, without straining. Very frequently, in cylindrical boilers especially, steam is raised by lighting the fire in one furnace only in each boiler: this will tend to set up circulation in the water and promote uniformity of temperature throughout the mass. When the steam pressure has reached a sufficiently high point for the purpose, the water in the boiler should be blown down to the ordinary working level, which will assist the circulation and prevent much of the racking occasioned by unequal

temperatures of those portions comprised in the water spaces of the boilers.

The safety- and stop-valve boxes should be kept properly drained, to prevent accumulation of water either from condensation when raising steam, or from water coming down the waste-steam pipes. This point should always be carefully attended to, especially before opening the stop-valves of any boilers in which steam is up. Serious accidents have in some instances resulted from the neglect of this precaution.

Funnel.—If the ship is fitted with a telescopic funnel, the cover must be taken off, and the funnel raised to its full height and secured in position, before lighting fires. The funnel-stays should be left sufficiently slack to allow for the expansion of the funnel, as its temperature becomes increased by the passage of hot gases through it.

The Kingston valve and sea-cock on the suction-pipe of the auxiliary feed-engine should be opened, and the pumps tested, as soon as the steam pressure in the boilers is sufficient for the purpose.

Density of water in the boilers.—It is the usual practice, when a ship is under steam, to test the water in the boilers by the hydrometer every hour, to ascertain its density. No brining or blowing-out is necessary until the density reaches 25 degrees, or 2½ times the density of sea water. The maximum working density is generally about 30 degrees for boilers fed with sea water only, and 40 degrees for those fed with water from surface condensers. The selection of the most suitable density in any particular case must depend on the experience gained as to the economy of fuel when under steam, and of the condition of the boilers on the examinations after steaming. As a rule, the bottom blow-out cocks should be used for keeping the water from exceeding the limits of density, and the feed-valves should be kept shut when blowing out,

to insure the water from the bottom of the boiler being expelled. It is necessary to open the surface blow-out or brine-cocks occasionally, to get rid of impurities floating on the surface of the water.

The deposition of scale on the surfaces of the boiler will not depend so much on the density at which the water is kept as on the quantity of sea-water that is passed into the boilers. The reduction of the loss from blowing-out by working at the higher densities, when sea-water is used as feed, was pointed out in Chapter XII. It is only reasonable, therefore, to infer that the deposition of scale would be less at the higher than at the lower densities, as less feed-water is required, and this appears to be confirmed by the experiments made by the Admiralty Boiler Committee. In their last report, dated March 25, 1880, they recommend that the maximum working densities should be increased to 35° for jet injection and 45° for surface condensation. In the latter case, in which the boilers are fed with fresh water, excepting the comparatively small quantity required to make up waste, the density indicated by the hydrometer would appear to be of little or no importance, and, as a rule, blowing out is only necessary to get rid of grease or other impurities.

Change of water.—In order to exclude air, in the absence of which oxidation cannot proceed, it is desirable to keep the water in the boilers without change, for as long a period as practicable, whether the fires are alight or not. The boilers should only be emptied when necessary for examination, cleaning, or repair. This will require to be done much oftener with boilers fed with sea-water than with those fed from surface condensers, as in the former case scale is continually being deposited on the heating surfaces while the boilers are at work. Experience as to the condition of the boilers on the examinations can be the only guide as to the greatest amount of steaming to

which it is safe to expose these boilers without allowing too thick a deposit to be formed on the heating surfaces. It is, however, clear that the higher the density at which they are worked the longer they can be kept going without the necessity of opening out for scaling, etc. With surface condensation the water need only be changed when it is dirty or acid, or when the boiler is required to be used for distilling; and these boilers need only be emptied at considerable intervals for examination, cleaning, or repair.

If necessary to reduce the density, or to prevent blowing-off from the safety-valves, cold sea-water, if clean, may be pumped into the boilers by the auxiliary feed-pumps; but this should be avoided as much as possible, because the air, gases, and salts held in solution are detrimental, both with respect to preservation and cleanliness. When the engines are done with, the safety-valves may be lifted a little by hand, to allow the steam to blow off, until the pressure is reduced to four or five pounds per square inch, when the valves and connections should be closed up to prevent admission of air, and kept so as far as possible while the water remains in the boiler.

In modern high-pressure marine boilers, the principal danger to be guarded against is the mineral oil deposit on the furnaces and combustion-chambers. The oil used for the lubrication of the cylinders and slide-valves is carried over into the boilers, and sometimes forms a black carbonaceous scale which is an almost perfect non-conductor of heat even when not more than $\frac{1}{16}$-inch thick. The quantity of oil used in the cylinders should therefore be reduced to the minimum absolutely necessary. In recent ships, sight-feed lubricators have been fitted, which enable the engineer to see every drop of oil that passes into the engines, and to regulate the quantity accordingly. This has considerably facilitated the reduction of the supply

of oil for internal lubrication, and has so far been beneficial to the boilers.

Chemical tests.—It is necessary that the water in the boiler should be prevented from becoming acid by decomposition of the lubricants or other matters. If it be kept in a neutral or alkaline state it will possess no corrosive properties. In order to ascertain the chemical condition of the water as to acidity, neutrality, or alkalinity, a small quantity should be drawn off every day when the boilers are at work, and tested with litmus-paper. If the water be found to be acid, it may be neutralised by putting a little common soda in the condenser or hot-well, from whence it will be pumped into the boilers with the feed-water. The use of milk of lime, which was described in Chapter VIII., appears to have the effect of keeping the water in the boilers alkaline.

Uniformity of temperature and pressure.—Every precaution should be taken to prevent sudden changes of temperature and pressure during the working of the boilers: the smoke-box doors should not be opened suddenly, as the rush of cold air through the tubes affects the ends, which, being so much thinner than the tube-plates, are liable to shrink and cause leakage. When it is necessary to check the generation of steam, as, for example, on easing or stopping the engines, the ashpit dampers should be closed and the steam pressure, if possible, prevented from rising too high by using the silent blow-off, easing the safety-valves, or opening the separator discharge-valve when it communicates directly with the sea, the smoke-box doors only being opened when absolutely necessary, and then not suddenly but gradually. The frequent opening of smoke-box doors to check the generation of steam is a convenient expedient, but one that cannot be too strongly condemned, as it inevitably causes injury to the boilers.

The boilers should never be emptied by blowing-out except in cases of extreme urgency, as this practice produces leaky tubes and joints. When steam is no longer required, everything on the boilers should be closed up, the fires allowed to burn out, and the water to become cold gradually, so that the whole of the boilers may contract uniformly, and prevent undue strains being brought on any part. When all is cold, the ashes and clinkers, etc., may be drawn out of the furnaces, and any fuel not entirely consumed should be saved and utilised for distilling purposes. The water may be pumped out from the boilers when cold, if necessary for examination, cleaning, or repair of the boilers. As pointed out in Chapter VIII., the practice of drawing fires shortly after the engines are stopped is objectionable, as the cold air entering the furnaces is liable to cause leakage at the fire-box ends of the tubes and in the joints of the combustion-chambers and backs of the furnaces.

Economy of fuel.—The greatest care requires to be exercised, in the working of the boilers, to promote economy in the expenditure of coal. The fires should be stoked carefully and regularly, the steam pressure and water level kept constant, and no waste should be allowed by steam blowing off from the safety-valves, which should always be kept tight to prevent leakage. The cinders and small coal that fall into the ashpits should be reburnt, and the fires should not be unnecessarily forced or disturbed. The firebars should be kept uniformly covered with coal of moderate thickness, especially at the backs and wings of the furnaces, to prevent cold air passing through to the combustion-chambers. In many cases in which reduced powers only are required, it will be found to be advantageous and economical to brick over a portion of the back lengths of the firebars. The wing firebars should be placed close to the sides of the furnaces, to

prevent the fire being too fierce close to the plates. If there are stay-nuts or other projections in the line of fire-bars, the wing-bars should be cast with recesses to fit over them, so that they may be quite close to the plates. In corrugated furnaces the wing-bars should be cast to fit the corrugations.

Economy of fuel may be promoted by careful stoking, attention to the admission of air for combustion, and by keeping the water level and steam pressure constant, the latter being a few pounds below the maximum load on the safety-valves, to prevent loss from blowing off. At reduced speeds, as pointed out at p. 169, the pressure in the boilers need only be kept a few pounds above the necessary initial pressure in the cylinders for the required speed of the engines. It is important to ascertain the most economical rate of consumption in the boilers per square foot of firegrate, so that when proceeding at slow speeds neither too many nor too few fires may be alight. In the former case too great a quantity of air will pass unconsumed through the fires, and carry off heat up the funnel, and in the latter there will be waste by unnecessarily forcing the fires.

Priming.—The passage of water with the steam from the boilers to the engines is technically called *priming*, and is one of the most inconvenient and disagreeable incidents that can occur in the engine department when the ship is under steam; and when it is violent, unless proper precautions be taken, may lead to serious and even dangerous consequences. It usually results from insufficient steam space or area of water surface in the boilers, especially when the fires are being forced, the ebullition being so violent that particles of water are carried into the steam-pipes with the steam. It may also be produced by opening the stop-valves quickly, which, by drawing off a large quantity of steam

from the boiler, and thus reducing its pressure, causes a rapid evaporation of the water in the boiler, the temperature of which is higher than that due to the suddenly-reduced pressure. With surface condensation the mineral oil used for the lubrication of the slides, cylinders, etc., is carried through the condenser to the boilers, and floats on the surface of the water, and if allowed to accumulate to too great an extent, may cause priming to occur. This may be prevented by judicious use of the surface blow-out cocks or valves. With cylindrical boilers especially, if the water level in any boiler should be allowed to rise too high when working at full power, the practical reduction of the steam space in that boiler will almost inevitably produce priming.

If the priming be excessive, water will rapidly accumulate in the separator and pass on to the cylinder, where its presence will be made known by the thumping of the piston at each end of its stroke in forcing out the water through the escape-valves. If the priming be slight, so that water does not accumulate to any great extent in the cylinders, but only causes the steam to be wet instead of saturated, its effect will be to reduce the speed of the engines, because the expansive force of the steam will be decreased; and the water carried over with the steam will evaporate during the period of exhaust, increasing the back pressure and diminishing the vacuum in the condensers.

The existence of priming is always indicated by the reduction of the vacuum in the condensers and the decrease in the speed of the engines. If it be excessive, it will be further indicated by the accumulation of water in the separator and cylinders, and by the agitation shown in the upper portions of the gauge-glasses on the boilers, caused by the mixing of water to a considerable extent with the steam.

When the priming is excessive, the relief-cocks on the cylinders should be opened, and the engines eased. The separator drain-cock should be opened to blow the water that accumulates in it overboard. The generation of steam in the priming boilers should also be checked by closing the ashpit dampers, and the scum-cocks or valves on the boilers should be opened to blow out the impurities from the surface of the water and reduce the agitation. If necessary, the donkey feed-pump should be started to pump cold water into the boilers.

It is a very injudicious practice to put the engines quickly from slow to full speed, as the sudden opening of the valves and increase of speed of the engines is very likely to produce violent priming, which, by the rapid and unnecessary withdrawal of a large quantity of water from the boilers, may cause the combustion-chambers and tubes to become overheated; and the sudden accumulation of water in the cylinders may cause them to be cracked by the blows of the water in front of the pistons, if it should be too great to be relieved with sufficient readiness by the action of the cylinder escape-valves. Engines should always be worked up gradually to full speed, so as to enable the rate of generation of steam in the boilers to keep pace with the demands of the engines.

Boiler tests.—To promote safety in the working of the boilers, and to serve as a guide for the reduction of the load on the safety-valves when it may become necessary, it is desirable that the boilers should be tested by water pressure at regular intervals. To carry out this test, the boilers are first filled, through the blow-out cocks, and the pressure is then produced by pumping additional water into them by means of the hand-pump in the engine-room. A special plunger, of small diameter, with suitable glands and bushes, is usually provided for the hand-pump, to

enable the necessary testing pressure to be produced. The thickness of the plates should also be ascertained occasionally by drilling small holes through them. The test-holes are afterwards tapped and filled with screw bolts.

During the application of the water-pressure, the boiler should be carefully examined and proper gauges used to detect any change of form in the furnaces and combustion-chambers. Deflection meters are now supplied to all ships for this purpose. The water-pressure test should be double the working steam-pressure, provided that during the examination no indication of weakness is observed. Should, however, any sign of probable permanent deformation be detected, the test should be stopped, and the load on the safety-valves should then be reduced to one-third of the test-pressure arrived at before such indication was seen, unless the defective part can be strengthened. If the drill test should show unusual thinness in any part, the water-pressure should be very carefully applied to prevent injury being caused from over-pressure.

Distilling.—When it is necessary to distil water for drinking, washing, and culinary purposes, it is desirable that, if possible, a boiler should be used that is not supplying steam to the engines at the same time; and care should be taken to insure that nothing but clean sea-water is pumped into this boiler while so employed, in order that none of the lubricating oil taken by the feed-water from the surface condenser may be carried into the distilling apparatus. If the boiler had previously been used for steaming purposes, the surface blow-out should be freely used before distilling, in order to get rid of any greasy matter that may be floating on the surface of the water. The pressure of steam used for distilling need only be just sufficient to work the donkey feed-pump and the small steam pump employed for pumping the

water from the distilling condensers to the tanks in the hold, say from 12 to 15 pounds per square inch. In some ships an auxiliary boiler is fitted, to be used for distilling purposes.

Boiler tubes.—When under steam, the boiler tubes should be swept and cleaned at regular intervals in order to prevent the accumulation of soot, the engines being eased for the purpose if necessary. In practice the tubes of only a portion of the boilers are cleaned at one time, generally during the middle watch at night, and this can be effected without any considerable reduction of speed. A similar arrangement is adopted for clearing the fires of clinkers, etc., when they become dirty. These processes should be completed rapidly, to avoid the passage of cold air through the furnaces and tubes, etc., so far as possible. Shortly after the fires are drawn, the tubes, and all other parts exposed to the fire, should be thoroughly swept and cleaned. Valves, with flexible hoses and suitable nozzles, are fitted to enable the tubes to be cleaned by steam.

When the ends of the tubes begin to leak they may be made tight by expanding them with a roller mandril, and when they become thin, ferrules, about one inch long, should be placed inside them and expanded by a mandril, so as to be quite tight and protect the thin end of the tube itself. These ferrules are usually made of malleable cast-iron.

Warming the engines.—As soon as the steam pressure in the boilers rises to a few pounds per square inch above the atmosphere, the stop-valves should be opened slightly to allow the steam to pass into the cylinders and jackets, to warm them and bring them gradually up to the working temperature before any strain is brought on them. The drain-cocks on the jackets and cylinders, etc., should be opened to allow the condensed steam to pass into the bilge. If the full pressure of steam were admitted to the

cylinders before they had been thoroughly warmed, there would be great danger of their cracking from unequal expansion of the several parts, especially in the case of large engines. The slide-valves should be worked to and fro from their central positions by means of the starting-gear, to allow free access of the steam to both sides of the pistons. The auxiliary starting-valves, when they are fitted, should also be worked during the warming of the engines. A considerable time should be occupied in the process of warming the cylinders, etc., to enable all the parts to expand slowly and uniformly. This point is one of great importance in preparing the engines for steaming.

As soon as practicable, the inlet and outlet valves to the surface condensers should be opened, and the circulating engines and pumps started, to cause circulation of cold water through the condensers, and prevent them from becoming heated by the drainage of steam from the cylinders during the process of warming.

If the circulating engines exhaust only into the condensers, they should not be kept at work more than is absolutely necessary, to prevent accumulation of vapour in the condensers, as, in general, the air-pumps, being attached to the main engines, are inoperative until the engines are at work.

During the process of warming, all the lubricators should be filled with oil, and the worsteds prepared to be put in place immediately before starting the engines; the gland of the stern-tube should also be slacked back and the hand turning-gear taken out and secured. These operations should on no account be overlooked. The steam fog-whistle, and the engine-room telegraphs and voice-pipes should also be examined and tested. Careful examination should be made to insure that no tools or any other articles remain in places where they would interfere in any way with the working of the engines.

As soon as the steam pressure is sufficiently high, and the engines have been thoroughly warmed, they should be worked a few revolutions ahead and astern, alternately, to insure that they are under control before weighing anchor or slipping moorings. This trial should not be sufficient to produce any way on the ship, but only to ascertain if the engines can be readily handled. With the present working pressures of steam a vacuum in the condensers is not generally necessary for starting the engines; but it is very important, to facilitate handling, that the cylinders should be thoroughly warmed and kept effectually drained of water; and in every case, before starting, even if the engines have only been at rest for a short time, the precaution of clearing the cylinders of water should be observed. Before starting, all the auxiliary engines, especially the feed-donkey, should be tested.

The engines should be started slowly, and gradually worked up to the required speed, so that the strains on the boilers and engines may be brought on uniformly and regularly. The water condensed in the cylinder-jackets during the working of the engines should be blown into the hot-wells or feed-tanks, and not allowed to accumulate in the jackets, which would impair their efficiency.

Slide-valve packing-rings.—It is necessary that the packing-rings at the backs of the slide-valves should be properly adjusted, to insure their being steam-tight without producing too great a pressure on the working faces. If they be slack, the equilibrium of the valve is destroyed, which increases the strains on the eccentrics and slide-rods and gear, and a considerable loss of efficiency will ensue from the passage of steam to the condenser from the back of the valve without performing any useful work. If they be too tight, the friction is increased, and unnecessary and unknown strains are brought on the eccentrics

and slide-gear. Particular care is required when india-rubber or elastic core packing is used, because of its liability to swell when exposed to the action of mineral oil.

Pistons.—Great waste of steam, and consequently of fuel, will ensue if the pistons leak and allow steam to pass from the steam to the exhaust side of the piston, and thus, in the case of the low-pressure cylinder, to the condenser, without the performance of any work. The metallic packing-ring should, therefore, always be kept pressed against the working surface of the cylinder. Advantage should be taken of every opportunity for taking off the junk-ring for inspection of the piston springs and renewal of any that may be found broken. In the case of horizontal engines, the pistons have a tendency to wear down and allow the steam to pass over them to the condenser. They should, therefore, be lined up to the central position, and the springs adjusted as required; and the metallic packing-rings should be turned some distance round as they become worn. When the pistons are fitted with back supporting-rods or trunks, the guides should be lined up and adjusted as required, to prevent as far as possible the weight of the piston from resting on the bottom of the cylinder.

Condensers.—The ends of the condenser tubes should be kept tight, new packings being fitted as required, to prevent the admixture of sea water with the feed. This is a point of much importance with respect to economy of fuel, and also as regards the preservation and cleanliness of the boilers. The water in the hot-well should be tested occasionally, to ascertain whether or not it is fresh. If the condenser tubes are found to be leaking when the engines are under way, care should be taken to prevent the water passing from the condensers to the cylinders when the engines are standing, and the ends of the defective tubes should be repacked on the first convenient

opportunity. Every precaution should be taken to prevent loss of fresh water from the hot-wells or feed-tanks. The tubes should be frequently examined and kept as clean as possible to increase their efficiency.

When the engines are not at work, the sea-cocks of the circulating-pumps should be kept shut and the condenser casings drained, to prevent corrosion. If the engines are not likely to be used for some time, the covers and doors of the foot and delivery valve chambers should be taken off, and all the condensed water drained out of the condensers and hot-wells, which should then be kept quite dry. The covers and doors need not be rejointed until it is necessary to again prepare for steaming.

The action of the mineral oil used for internal lubrication on the india-rubber valves of surface condensers causes them to swel in a peculiar manner, and soon destroys them, unless care and watchfulness be exercised. They require to be frequently examined, and cut when necessary to prevent their overlapping each other. Their durability may be extended by occasionally washing them in a solution of caustic soda or potash to remove any greasy matter that may be adhering to them.

Leakage through the stuffing-boxes of the rods passing through the cylinders and air-pumps and on the eduction pipes should be guarded against, as the admission of air through these fittings would impair the vacuum in the condensers and reduce the efficiency of the engines. The nuts on the glands of the rods are usually fitted with toothed or worm gearing to enable them, if necessary, to be screwed up equally while the engines are at work. The glands of all the cocks, pipes, etc. in connection with the condenser should be kept quite tight, to enable the vacuum to approach as nearly as possible to the maximum attainable with the working temperature of the condenser.

Water service.—The bearing surfaces should be so

designed and arranged that, when properly adjusted, the ordinary lubricating arrangements may be sufficient to keep the journals, etc. from heating when the engines are being worked at full power. To provide, however, for the contingency of faulty adjustment, dirt getting into the bearings, or the friction being temporarily increased from any other cause, small pipes with stopcocks are led from one of the sea-cocks to each of the bearings, to enable cold water to be run on them in case of their overheating. These should, however, be used as rarely as possible, as the water tends to corrode and destroy the bearing surfaces. If in any case it may be necessary to use water on the bearings during the working of the engines, the water supply should be discontinued for some time before stopping, and oil only used instead, so that the journals may become coated with oil and preserved as far as possible from rusting after the engines are stopped. Caution is necessary in the application of cold water to heated bearings, to avoid injury to the brasses from sudden contraction. In the case of hot bearings it is often advantageous to add a little powdered blacklead or sulphur to the oil, to assist in carrying off the heat generated by the friction. The blacklead or sulphur should be sifted through bunting, to cause it to be quite free from lumps and grit.

Crank-shaft journals.—It is very important that the crank-pin and crank-shaft bearings should be kept properly adjusted, and that the engines should not be allowed to work with the journals slack. This would leave the shaft improperly supported, and the hammering of the pistons on the crank-pins at each stroke would bring severe bending strains on the crank-shaft. It is probably to this action more than any other that most broken crank-shafts are due, as the strains are thereby enormously increased, and, from the nature of the case, it is difficult to reduce it to exact calculation. In all cases the main-

bearing and connecting-rod brasses should be screwed tightly on to the stops or liners, and not left loose.

Economy.—All the appliances fitted for economising fuel, such as steam-jackets, expansion-valves, etc., should always be used, except under special circumstances; and it is generally desirable, when working at reduced powers especially, to raise the link to a considerable extent, so as to reduce as far as possible the loss from the clearance spaces in the cylinders. This, in compound engines, also has the tendency of equalising the work done by the two cylinders, as pointed out on p. 343.

When the ship arrives in harbour and the engines are finally done with, the regulating- and stop-valves should be closed, the worsteds taken out from the lubricators, cylinder and jacket drains opened, and the engines wiped down while warm to clean off the grease, etc. adhering to them. All sea connections should be closed, and the bilges pumped out by one of the auxiliary engines before the steam pressure in the boilers is too far reduced.

If the ship has been for a considerable period under steam, and the engines are not likely to be again required for some time, advantage should be taken of the interval to make a thorough examination and adjustment of the working parts and to remedy all defects. The cylinder covers or manhole doors should be removed and the junk-rings of the pistons taken off for examination of the springs, etc. The pistons and the insides of the cylinders should be cleaned and oiled, and the pistons reconnected. The manhole doors should not be rejointed, so that the internal parts may be kept clean and free from rust until again required for steaming. The slide-bonnets should be removed, slide-valves examined, and taken out if necessary, the surfaces cleaned and oiled, defects, if any, made good, packing-rings adjusted or refitted as required, and covers rejointed. The packings of all the glands, especially

those on the cylinders and condensers, should be examined, and renewed where necessary. The air-pump covers and condenser doors should be taken off for examination of the packings at the ends of the tubes, and of the india-rubber foot and delivery valves. The insides of the condensers and hot-wells should be dried, and scraped and painted if required. The covers should be replaced but not rejointed.

All the bearing surfaces, working parts, and fastenings should be carefully examined and adjusted, the connecting-rod and main-bearing brasses being *let together* as may be found necessary. The bolts of the bearings should be drawn back and thoroughly cleaned and coated with oil or tallow before being replaced, so that no corrosion may ensue in consequence of the use of water on the bearings. The coupling-bolts of the screw shafting, especially those in the driving and disconnecting couplings, should also be carefully examined, and some of them drawn back to insure that they are in good order.

The boilers and boiler mountings, so far as may be practicable, should be examined and all defects made good, so as to put all parts in good and efficient working order. All the auxiliary machinery and gear should also be examined, properly adjusted, and repaired if required.

During the time the ship remains in harbour after the examinations have been completed and defects made good, all the engines, main and auxiliary, with the gear attached to them, should be kept clean and oiled, and partly turned round every day by the hand turning-gear. The slide-valves of each of the engines should be worked daily by the starting-gear, and levers and other working parts moved occasionally to prevent them from sticking. The sea and bilge cocks or valves of the pumps should be opened and closed daily, and the watertight doors, sluice-valves, Kingston, flooding, and other sea-cocks and valves, and the cocks or valves on the fire-main, should be worked

regularly every week to insure their being kept in proper working order.

Before getting up steam the undermentioned parts of the machinery, with the gear attached to them, should be examined, and, if movable, worked, to ascertain that they are in good order:

1. Main and auxiliary stop-valves.
2. Safety and sentinel valves.
3. Brine and blow-out cocks or valves, pipes and Kingston valves.
4. Steam gauges and water-gauge cocks.
5. Feed-cocks, valves and pipes, and all other boiler mountings.
6. Steam-pipes, expansion and other joints.
7. Stop, throttle, expansion, and slide valves.
8. Auxiliary starting-valves, and starting-gear.
9. Cylinder escape-valves and relief-cocks, and all jacket or other drain cocks or valves.
10. Sea-suction and discharge valves and cocks in connection with the surface condensers, main-engine pumps, and auxiliary engines.
11. The shaft couplings, nuts, cotters, and keys connecting the working parts, and all other fastenings of the machinery, should be sounded and overhauled. The pins in the driving and disconnecting couplings should be drawn back to ascertain that they are free in the holes.

Coal-bunkers.—Precautions have to be taken to prevent accident from explosion of gas in the coal-bunkers. The coal-shoots should always be kept quite clear of coals to permit the gas to escape through the grated covers on the deck. Ventilating-pipes are usually carried from the upper parts of the bunkers to the funnel casings, to allow the impure air and gases as they form to pass away freely to the atmosphere. The inlet-pipes are also led to the upper parts of the bunkers, as far as possible from the

outlet orifices, so that the ventilation may be from the surface and not through the body of the coal. The ventilating-pipes from the several bunkers should be independent of each other. The bunker lids should also be taken off at intervals to keep the bunkers well ventilated. Moisture sometimes causes a rapid generation of heat and gas, especially when the coal contains a considerable quantity of pyrites, and care must be taken to prevent wet coal being shipped and to insure the coal being kept as dry as possible after it is placed in the bunkers. Ships should not be coaled on rainy days if it can be avoided. The close bunker lids should be replaced as soon as possible after coaling to prevent water passing into the bunkers when the decks are being washed. Every precaution should be taken to ventilate the bunkers, and they should be tested by using safety lamps before sending men to work in them. Whenever the coal-bunker lids are removed, lights should not be brought near the openings until the accumulated gas has been allowed to escape. Small tubes with screwed deck-plates are fitted in the bunkers at regular intervals, about ten feet apart, to enable the temperature to be ascertained. This should be done regularly and frequently, and if the temperature should be found to be increasing, suitable means should be adopted for ventilating the bunkers and getting rid of the gas.

CHAPTER XXVIII.

MATERIALS USED IN CONSTRUCTION.

THE material that is most extensively used in the construction of engines and boilers is iron; using the term in its most general and inclusive sense to comprise the three forms in which it exists, viz. cast-iron, wrought-iron, and steel; which, though differing so greatly in appearance and qualities, are, in reality, but different forms of the same material.

Iron is very rarely found in the metallic state, but is generally combined with oxygen and carbonic acid, and mixed to a greater or less extent with clay and earthy matters. In this condition it is called *iron ore.* The most common ores of iron are:

1. *Magnetic iron ore*, which consists of magnetic oxide of iron, pure or nearly so, and contains about 72 per cent. of iron. From this ore some of the purest qualities of iron are produced.

2. *Red iron ore.*—This is peroxide of iron, pure or mixed. When pure and crystalline, it is called *specular iron ore*, or *iron-glance.* When in kidney-shaped masses showing a fibrous structure, it is called *red hæmatite*; when mixed with clay and sand, it is called *red ironstone* and *red ochre.* The pure ore contains about 70 per cent. of iron.

3. *Brown iron ore.*—This ore contains the same oxide of iron as the red ore, but in this case it is in a

state of chemical combination with water, the latter varying from ten to fourteen parts in a hundred parts of the ore. It may therefore be described as hydrated peroxide of iron. When compact and nearly pure, it is called *brown hœmatite*, which contains about 60 per cent. of iron; when earthy and mixed with much clay, *yellow ochre*.

4. *Carbonate of iron*.—This, when pure and crystalline, is known as *sparry* or *spathic iron ore*. When it is mixed with clay and sand, it is called *clay ironstone*. In many cases the clay ironstone is coloured black by the admixture of bituminous and coaly matters: when in this condition, it is known as *black ironstone* or *black-band ore*. The proportion of earthy matter in the ordinary ores containing carbonate of iron is from 10 to 40 per cent. The percentage of iron in the pure ore varies from 17 to 50 per cent.

British iron is made partly from red hæmatite, but principally from clay ironstone, and black-band. The red hæmatite furnishes iron of the best quality; but this ore is difficult to fuse by itself, so that it is customary to mix the lighter and inferior ores with it, which detract to some extent from the quality of the iron produced. It is not within the province of the present treatise to describe the details of the various processes adopted for extracting the iron from its ores. These are explained in all the well-known text-books on metallurgy. The principles involved in the reduction of the ore may, however, be briefly stated in general terms as follows:

If the iron be in the form of a carbonate, the carbonic acid is liberated by the heat, leaving oxide of iron; the oxygen from the air unites with the carbon of the fuel, forming carbonic acid, which gives up half its oxygen to the red-hot carbon and is converted into carbonic oxide. The oxide of iron is reduced to metallic iron by giving up

its oxygen to the heated carbonic oxide, thus again converting the gas to carbonic acid. The metallic iron unites with a small percentage of carbon to form *cast-iron*. The earthy constituents of the ore are removed by means of the chemical affinity of other earths, especially lime, forming an easily fusible glassy substance called *slag*.

The substances employed in smelting are:

1. The ore itself.

2. The fuel, which produces heat by its combustion, and supplies carbon.

3. The air, which supplies oxygen for the combustion of the fuel and for combination with the carbon in the ore.

4. The flux, generally lime, which promotes fusion of the ore, and combines with the earthy portions of the ore, forming a slag.

Cast-iron.—The iron, after it is reduced from the ore, is drawn off from the blast-furnaces, run into a series of shallow gutters or grooves, and broken into short pieces, say about 2 or 3 feet long. It contains in its composition a proportion of carbon, from 3 to 5 per cent., and is known as *pig-iron* or *cast-iron*. Only a part of this carbon is actually in chemical combination with the iron, say from 1 to about 2½ per cent., the remainder being diffused throughout the mass in the form of graphite or plumbago.

Spiegel-eisen.—Manganese is present to a greater or less extent in all varieties of cast-iron. Its influence is to prevent the separation of the graphite on cooling, and consequently to favour the production of the whiter and harder varieties of cast-iron. The cast-iron from the spathic iron ores contains a particularly large quantity of manganese, sometimes exceeding one-tenth the weight of the cast-iron. This iron is capable of retaining more than four per cent. of carbon in chemical combination, and is called *spiegel-eisen*, or *specular iron*. It is largely

employed in the Bessemer process for making steel, and enables the proper proportion of carbon to be added to the molten iron in the *converter*, with great facility and exactness.

From its low first cost, its strength, and the facility with which it can be cast into any form, *cast-iron* is extensively used in all engineering work. In the marine steam-engine, the cylinders, pistons, slide and expansion casings and valves, condensers and hot-wells, framing, plummer blocks, and many other parts, especially those of intricate form, are generally made of cast iron. In land engines the stop- and safety-valve boxes and steam-pipes are also usually made of cast iron.

The properties of different brands of cast-iron vary very widely according to the quality of the ore from which they are produced, and to the proportion of carbon actually combined with the iron. The iron that contains the greater quantity of carbon in combination is called *white cast-iron* from the appearance of the fracture. It is very hard and brittle, and unsuitable by itself for foundry purposes. At the other end of the scale is *grey cast-iron*, in which the greater proportion of the carbon is diffused throughout the mass in small particles of blacklead or graphite, which give to the fracture a greyish colour. This is much softer and tougher than the white iron, and is generally used in making castings, being mixed with some of the whiter varieties, or with good scrap cast-iron, to give it sufficient strength and hardness for various purposes. Cast-iron with properties intermediate between those of white and grey iron is often called *mottled cast-iron.*

Cast-iron is improved in strength and closeness of texture by remelting in the foundry cupola, the quantity of uncombined carbon being thereby reduced. The grey pig-iron as received from the blast furnaces is not alto-

gether suitable, alone, for engine castings, especially in cases where hard and smooth working surfaces are required. In such cases it is desirable to mix with it a proportion of good scrap cast-iron, from old engine castings, to increase both the strength and uniformity of the casting. The proportion of scrap varies from 30 to 70 per cent. according to the degree of hardness required.

Chilled castings.—If molten grey cast-iron be cooled rapidly when poured into a mould, some of the free carbon will combine with the iron, and white cast-iron will be formed. This property is taken advantage of in cases where any particular portion of a casting is required to be made harder than the rest, as, for example, the points of iron shot and shell. In such cases, the mould at the part where the casting is required to be hard, is made of iron or steel, so that the heat will be conducted away rapidly. Such castings are usually known as *chilled castings.* The depth to which the iron will be affected by the chilling will vary with the original quality of the iron used and the arrangements adopted for carrying off the heat. In all castings the outermost layer, which is in contact with the sand of the mould, will cool more rapidly than the other parts, and consequently the outer surface, or skin, of the casting will be harder and stronger than the parts nearer the centre. This hard skin should not be cut away more than can be avoided in castings that have to sustain severe strains.

Unequal contraction in cooling.—The great objection to the use of cast-iron, especially for parts that have to sustain severe and intermittent strains, is the uncertainty that exists as to its actual strength in any particular instance, in consequence of the unequal and unknown stresses brought on the material during the process of cooling in the mould. This initial strain is sometimes so great that the casting is found fractured on being taken

out of the mould, before it has been subjected to the action of any external force, and it is clear that it must in any case weaken the casting. The amount of contraction varies with the size and thickness of the casting, and with the quality of iron used. In thin castings the contraction is about $\frac{1}{16}$ inch in a foot, whilst in thick castings it is as much as $\frac{1}{8}$ inch per foot.

To prevent unequal contraction to as great an extent as possible, sudden variations in the thickness of the several parts should be guarded against, and the metal should, if practicable, be brought up by a hollow sweep from the thin to the thick section. It is in many cases desirable to do this in the mould, even if the rounded part has afterwards to be cut away by tools to form the finished article. Sharp corners should be avoided, as, in cooling, the crystals arrange themselves normally to the surface, which would leave a plane of weakness at the corner. It is advisable to arrange in the design for all the corners to be well rounded. Suitable arrangements should also be made to cause the rate of cooling of the different parts of the casting to be as nearly uniform as possible. Unless proper precautions are taken this unequal contraction will often cause distortion of form in castings of irregular shape, and it is therefore an advantage to make the castings as symmetrical in form and uniform in thickness as is consistent with the purposes for which they are required.

Cast-iron is also liable to have its strength reduced by the existence of *blow-holes* or gas-bubbles underneath the surface, which cannot generally be discovered by any ordinary inspection or test, if they should be at any depth below the surface. If near the skin of the iron, they may be discovered by tapping the casting with a hammer.

In consequence of these defects, cast-iron is an unreliable material for structures of irregular form that have

to sustain intermittent heavy loads, and a large margin of strength should be allowed when it is employed. It, however, has the advantages of cheapness and stiffness, and at present it is the only material that can be used for many parts of the machinery. If it can be avoided it should not be used in parts that have to withstand unequal and irregular temperatures; but in cases where the heat is uniformly distributed—such, for example, as in a steam cylinder when at work—it answers very well. It is, however, a treacherous and unreliable material to be used for parts that are exposed to unequal temperatures or subject to blows. For this reason, and not for want of strength, cast-iron is being discarded for stop- and safety-valve boxes of marine boilers. Several serious accidents have occurred from the bursting of cast-iron stop-valve boxes when steam, even of low pressures, has been admitted to them without previously taking the precaution of draining out the cold water that had accumulated in the valve-boxes and steam-pipes. These fittings are now usually made of gun-metal, to prevent accident from this cause.

The strength of cast-iron under the action of a crushing load is much greater than when exposed to tension, its resistance to crushing being from 80,000 to 110,000 pounds per square inch, whilst its average tenacity is only from 16,000 to 18,000 pounds per square inch. It is, therefore, in general, much more suitable for parts that are exposed to compression than for those that have to sustain stretching or tension.

The soundness and compactness of a casting is promoted by casting it under pressure. Consequently cylinders, pipes, etc. should be cast in a vertical position, with a *head* or additional column of metal above, whose weight serves to compress the mass of metal in the mould below. The dross and gas-bubbles ascend into the head, which is cut off when the casting is cool.

Malleable cast-iron.—By imbedding an iron casting in oxide of iron, or powdered red hæmatite, which consists almost entirely of peroxide of iron, and keeping it at a high temperature for a sufficient time, which will vary with the size of the casting, a portion of the carbon contained in the iron will unite with the oxygen in the oxide, and the casting will be converted, to a greater or less extent, into a material resembling mild steel or wrought-iron. This material is much cheaper than wrought-iron or steel; but the process is only applicable to comparatively small articles of fairly uniform thickness.

Wrought-iron.—Wrought or malleable iron, in its pure state, is simply metallic iron, without admixture or combination with any other element. The wrought-iron in general use, however, always contains impurities to a greater or less extent, some of which seriously affect its strength and working qualities. The substances that act most injuriously on the iron, if they are present in considerable quantities, are sulphur and phosphorus. Sulphur, in general, comes from the coal or coke used as fuel, and causes the iron to be *red-short*, or brittle when hot, so that it cannot be worked well at the forge. Phosphorus comes from phosphates of iron or lime either in the ore, the fuel, or the flux, and makes the iron *cold-short*, or brittle when cold, which is a more serious defect than *red-shortness*. It is also supposed that calcium and magnesium tend to make iron *red-short*, and silicon to make it *cold-short*. It is therefore very important that, in the reduction of the ore, proper steps should be taken to eliminate these impurities to as great an extent as possible.

The processes for producing wrought-iron from cast-iron, by refining and puddling, and of rolling iron into plates, etc., were described in Chapter VIII., and we now

proceed to mention a few of the more general properties possessed by wrought-iron, and its application in the construction of the several parts of the steam-engine.

Wrought-iron is the material that until recently was universally employed for the construction of the boilers, crank and propeller shafting, piston and connecting rods, and nearly all the moving parts of the engines, especially those that are subject to severe and varying strains. It is tough and strong, and has a fibrous structure, which renders its resistance to tension much greater than its resistance to compression. It is malleable and ductile, and though it can only be fused with difficulty, and at a very high temperature, it possesses the property of *welding*, when raised to a white heat (say 1500° to 1600° Fahr.), which enables two pieces of iron to be firmly united or welded together by hammering so as to form one piece. This very much increases and extends the usefulness of this material and enables it to be utilised for making large forgings.

Crank-shafts and other large iron forgings are made entirely by the process of welding iron under the steam-hammer. In the first place, small pieces of good scrap iron, such as cuttings and shearings of boiler-plate, etc., which have been previously cleaned in a revolving drum, are piled up to about 12 inches cube. These piles are heated in a reverberatory furnace to a welding heat and then hammered into slabs, called *blanks* or *blooms*, under the steam-hammer. The blanks are themselves afterwards piled or fagoted with the grain well crossed, heated in the furnace, welded into a solid block, and *drawn out* under the hammer; additional iron (blanks) being added and drawn out as required to form the forging. All important wrought-iron forgings for engine work should be *piled* or *built up* in this way so as to be perfectly sound and solid, and afterwards finished at the

smith's fire if necessary. The welding of rolled bars at the smith's fire, and *stumping-on* of projections, etc., though suitable and sufficient for many ordinary purposes, and cheaper than *making iron* and forging out of the solid, cannot be depended on with sufficient confidence for parts that have to sustain such severe strains as the working parts of steam-engines, and in which fracture would be attended with such serious consequences.

In engine forgings abrupt changes of form and dimensions should be avoided as far as possible, and great attention should be paid, in drawing out the material, to insure the continuity of the fibres near the surface; the processes of forging being conducted in such a manner as to cause the fibres to lie in layers parallel to the surface. To insure good welds and clean and sound forgings, it is important that the surfaces of the iron should be clean, and that there should be an entire absence of oxide, cinder, and dirt. Borax or sand is often thrown on the iron when at welding heat, which substances form a very fusible slag with any oxide or dirt on the iron and enable it to be readily squeezed out during the welding. Wrought-iron is at first made more compact and strong by reheating and hammering, but it soon reaches a state of maximum strength, after which any reheating or working rapidly reduces its strength and ductility.

The qualities of different brands of wrought-iron vary very considerably according to the amount of impurities contained and the care and skill exercised during manufacture. This is more noticeable as regards ductility and working qualities than with reference to the actual breaking strength. Many inferior brands have an ultimate tensile strength equal to that of the superior qualities of iron, but they are brittle, and will not stand working at the fires. When they are broken, the elongation is not more than about 6 to 8 per cent., whilst many of the

superior brands, such as the Yorkshire irons, will elongate as much as 20 to 25 per cent. before fracture ensues; and they are capable of being worked into almost any form, by a skilful workman, without fracture or deterioration of strength.

The fibrous structure which the best wrought-iron shows may be considered an abnormal state of the metal, being produced entirely by the processes of piling and hammering or rolling to which it is subjected in manufacture. Vibration tends to reduce the iron to its normal condition, which is crystalline, as is the case in all metals, and this probably accounts for the numerous fractures of shafts subject to vibration and intermittent strains, the fractures in most cases showing crystalline or granular surfaces. General experience shows that if wrought-iron shafts, such as railway axles, marine engine shafting, etc., which are exposed to intermittent strains, be designed with the ordinary factor of safety used for machinery, they are very liable to fracture after running for a certain period, and in all such cases the appearance of the iron at the fractured part has been crystalline. These parts are therefore generally made larger than at first sight appears necessary, so as to give increased stiffness and reduce the intensity of the torsional strains on the particles; and the durability of the shafts is thereby considerably extended. For similar reasons much of the shafting for land engines, in which weight is comparatively unimportant, is often made of cast-iron. In these cases the shafting has to be made much larger and heavier than if it had been made of wrought-iron, but since it is in its normally granular or crystalline condition, it does not become weaker by work; so that if it be made strong enough in the first instance it will still retain its original strength after many years' work, because its molecular condition is not altered by the vibration and strains to which it is subjected.

Case-hardening.—The outer skin in many wrought-iron pieces of machinery is made hard, to resist friction, by the process of case-hardening, which consists in imbedding the article in some carbonaceous substance, and raising it to a red heat, by which means the outer layers acquire sufficient carbon to convert them into steel. Generally speaking, substances which contain nitrogen as well as carbon are most suitable for this purpose. One of the most convenient is the yellow prussiate of potash, in the preparation of which, bone, horn, and other similar animal substances are indispensable. The iron is sprinkled with the finely-powdered prussiate of potash, heated to redness without access of air, and afterwards cooled in water. The depth of the hardening will depend on the time occupied in the process. In marine engines, the pins of the link motion, and many other similar parts, are usually made of iron case-hardened.

Steel.—The term steel is applied to all compounds of iron and carbon in which the proportion of combined carbon does not exceed 1·5 per cent. The properties of the materials thus included under a common name vary, however, very greatly, according to the amount of carbon they contain. When the percentage of carbon is below 0·5, the material is called *mild steel* and possesses few of the qualities popularly attached to steel, and in point of fact, very closely resembles the best wrought iron. The hardness and tenacity of steel, and its capability of fusion, increase as the percentage of carbon becomes greater.

Steel is produced in general either by the addition of carbon to wrought iron, or by the abstraction of carbon from cast-iron. The former method, although more complex and expensive, is preferred for making the higher classes of steel required for tools, etc., as wrought-iron can be obtained in a greater state of purity than cast-iron. The second method is employed for making large quantities

of steel rapidly and cheaply, such as that required for plates, bars, rails, &c.

Until a comparatively recent date all English steel was produced by the process of *cementation*. This consisted in imbedding bars of wrought iron in charcoal, and exposing them to a high temperature for a considerable period, during which time a portion of the carbon combined with the iron. The steel thus formed is called *blister steel*. If the bars of blister steel be cut up into lengths, piled or fagoted together, heated in a furnace to a welding heat and hammered, *shear steel* is formed. This steel does not possess sufficient hardness to be suitable for tools that require keen cutting-edges, but it is used for making large shears and other tools for rough work in which considerable toughness is necessary. *Cast steel*, which is the hardest, strongest, and most compact description of steel, and which contains the largest proportion of carbon, is produced by breaking the blister steel into small pieces and melting them in a crucible. This material is used for all cutting tools and instruments which require sharp and well-defined cutting-edges. It is very brittle at high temperatures, so that it can only be forged with difficulty, and is incapable of welding. By *Heath's process*, however—which simply consists of the addition to the steel in the crucible of about one-hundredth part of its weight of carburet of manganese—the tenacity at high temperatures of the steel produced is much increased, so that it may be forged and welded with much greater facility.

Steel is distinguished from wrought-iron by its capability of being cast into a malleable ingot, so that uniformity of structure may be insured; and also by its possessing the property of *tempering*, which enables it to be hardened by sudden cooling, or softened by gradual cooling from a high temperature. Its degree of hardness can be regulated with exactness, by suitably selecting the tempera-

ture from which it is suddenly cooled. The steels which contain the larger proportions of carbon, such as cast steel for example, possess this property of tempering in the highest degree, and this is most valuable in the case of cutting tools. The mild steel used for engine forgings and boiler plates, which only contains from 0·15 to 0·30 per cent. of carbon, appears to be almost, if not entirely, destitute of hardening qualities. It is difficult to give definite rules for the tempering of tools, as the proper degree of hardness for any particular case can only be determined by experience. The usual method adopted in practice for tempering tools, is to first make them red-hot, and then to harden the cutting-edges by cooling them suddenly in water, afterwards allowing their temperature to increase by the conduction of heat from the adjoining thicker portion of the tool, until it reaches the desired point, when the whole of the tool is cooled to fix or set the temper. The instant when this should be done will vary with the purpose for which the tool is required, and will be indicated by the colour shown by the thin film of oxide formed on the surface as its temperature increases. It will be at first a light straw colour, at which point the temper, if set, would be very hard. This is only used for light wood-cutting tools which require a very keen edge. The colour afterwards changes to a deeper yellow, then brownish-yellow, purple and blue. Tools for cutting brass require to be made harder than those for iron, and the lowest tempers are employed for chisels and other tools that are subject to rough usage. Most of the tools used in working metals require to have their tempers set whilst the steel is passing through the various shades of purple, and before it arrives at the blue stage.

Puddled steel.—This is produced by puddling pig-iron, the process being arrested when the proportion of carbon

remaining in the material is sufficient to form a mild kind of steel. The damper is then shut, and the puddle-balls treated in the ordinary manner. The puddle-bars are also *shingled* and worked into bars, plates, &c., as in the case of iron.

Bessemer steel.—In the Bessemer process for making steel, molten pig-iron is poured into a vessel called a *converter*, through which a stream of air is blown by a strong blast. The oxygen of the air first removes the silicon, and then unites with the carbon in the cast-iron and carries it away. After all the carbon has been removed, the proper proportion of carbon required to make the steel is introduced by the addition of the requisite quantity of molten *spiegel-eisen* to the iron remaining in the converter. The steel thus made is poured into ingots, and afterwards hammered, rolled, and worked as required.

Siemens-Martin steel.—In the *open-hearth* process of making steel, the great heat produced by a Siemens regenerative furnace is used for the purpose of dissolving, in a bath of molten pig-metal, ores of iron either in a raw state or in a more or less reduced condition, in order to reduce the quantity of carbon in the pig-iron, the oxygen in the ore uniting with the carbon to form carbonic oxide which passes off as gas. Usually steel scrap is used in addition to the iron ores. This not only utilizes the scrap which would otherwise accumulate on the premises, but also improves the quality of the material and enables the charges in the furnaces to be worked more quickly. The principal advantage of this system of producing mild steel is, that it is not dependent on a limited time for its results, as is the case in the Bessemer process. The heat of the furnace is such that the fluid bath of metal, after having been reduced to the lowest point of decarbonisation, may be maintained in that condition for any reasonable length of time, during which, samples may be taken and tested,

and such additions of pig-metal, steel scrap, or iron ore may be made to it as may be necessary to adjust it to the required quality, and uniformity of quality in all the ingots produced may thus be insured. In many cases, to improve the quality of the material, a small quantity (about one per cent.) of ferro-manganese is thoroughly mixed with the metal in the ladle before running it into the ingot mould.

Mild steel can be worked well at a red heat, and can be bent cold into most ordinary forms. It appears, however, to possess the peculiar property of becoming brittle at a temperature between about 400° and 600° Fahr.—which is technically known as a *blue heat*, from the colour of the fracture at that temperature. Care should therefore be taken to prevent any work being done on the material after it has fallen to this dangerous limit of temperature. All plates or bars that have had much work done on them while hot should be subsequently annealed. The annealing, if possible, should be performed simultaneously over the whole of the plate or bar in question, and care should be taken to prevent the access of air to the furnace, or the impinging of the flame on the material during the process. After the material has been raised gradually to a red heat, it should be taken out of the furnace and allowed to cool slowly. Annealing is not necessary in cases where the whole of the plate has been heated and bent or flanged at one heat.

Steel made by the Siemens-Martin process is now extensively used for engine forgings and for boiler-plates instead of wrought-iron. To insure soundness the ingots from which the plate or forging is made should be cast with a large head. During the solidification the lower part of the ingot has to feed itself from the head, forming a funnel-shaped cavity usually known as 'the pipe,' which is essential for the ingot to be sound. The head must be long enough to allow the whole of this pipe to be cut

away and the forging to be made from the sound metal below it.

Whitworth's fluid compressed steel.—Sir Joseph Whitworth's system of producing sound steel ingots, free from blow-holes and suitable for the best engine forgings, consists in subjecting the metal while setting in the ingot mould, to great hydraulic pressure, and by this method steel of very great uniformity and strength is produced. This has been extensively used for crank and propeller shafting, cylinder liners, and many other purposes. In the Whitworth system the whole of the forging is performed by suitable and powerful hydraulic presses, no hammering whatever being employed, but the ingot gradually squeezed to the required form. There can be little doubt that this system is superior to the use of steam hammers and it has recently been adopted by several other leading steelmakers.

Steel castings.—The most important development of the application of steel in the construction of modern engines has been its use in the form of mild steel castings for parts that were formerly made of cast or wrought iron. In many recent engines the frames, bedplates, pistons, screw-propellers, and many other parts have been made of mild cast steel, and this has enabled considerable reduction of weight to be effected. The greater the proportion of carbon contained in the steel the more easily will it flow in casting, and the less danger will there be of cavities or blow-holes. The highly carburized steels are, however, so hard and brittle that there is much difficulty in working them, and they are generally deficient in the toughness requisite for most parts of machinery. One difficulty in making mild steel castings, containing a low percentage of carbon, is to prevent the formation of gas-bubbles or blow-holes. At the Terre-Noire Steel-works in France, many experiments relative to the production of sound castings from

mild steel have been made, and it has been found that by the addition of a silicide of manganese to the metal just before casting, the blow-holes may be prevented, and the material produced possesses many remarkable qualities. The silicon in the alloy prevents blow-holes by decomposing the oxide of carbon, forming silica. The manganese reduces the oxide of iron, and prevents a further production of gases by the reaction of the oxide on the carbon. A fusible silicate of manganese and iron is consequently formed which passes away with the slag. A full description of the process is given in a paper read in 1877 at the Iron and Steel Institute by M. Gautier, of Paris.

This system of making mild steel castings is adopted by some English steelmakers, but others consider that the requisite composition to insure, not only strength and ductility, but also soundness and solidity, can be obtained by careful attention to the materials employed and to the details of the processes of manufacture.

The greatest difficulty to be overcome in making mild steel castings is to prevent 'draw' in the metal from the contraction while cooling in the mould. The steel requires a temperature of about 4,000° Fahr. to melt it, as compared with 2,000° Fahr. for cast-iron, and the contraction of the steel casting is as much as $\frac{5}{16}$-inch per foot. Steel castings cannot be 'fed' as in the case of iron ones, and consequently the head must be of sufficient size and suitably arranged to allow of this feeding action taking place, and the milder the steel the greater this will be. This forms the 'pipe' in the head, which is generally an index of a sound casting, for if the casting in contracting does not feed from the head, it must feed from itself and became unsound. All steel castings after being taken from the mould should, without first being allowed to cool, be reheated and annealed to insure molecular equilibrium and freedom from internal strains.

There can be little doubt that as further experience in manufacture is gained and more confidence felt as to the general soundness and uniformity of steel castings, their use will be greatly extended, not only to replace parts hitherto made of cast-iron or gun-metal but also many parts which are now made of forged iron or steel. In some cases piston and connecting rods, crank-shafts and propeller-shafting have been made of steel castings without any forging. In fact some steelmakers are of opinion that large shafts and other important parts of engines would be stronger and more durable if made of cast-steel, instead of being forged; this view has not yet, however, been generally accepted.

Mild steel castings have been used with much advantage to replace many large and complicated ship forgings, such as stems, stern-frames, rudders and tillers, outer brackets for propeller-shafting, &c., which, as forgings, are difficult and costly to make and are frequently unsafe in consequence of imperfect welds.

After iron and steel, the materials most generally used in the construction of the machinery are copper, tin, zinc, lead, and their alloys.

Copper.—This metal is red in colour and very soft, malleable, and ductile when cold. It cannot be welded and does not make good castings. It can, however, be readily worked cold, and it is consequently used principally for making steam and other pipes which require to be bent cold. For most other purposes it is too soft and weak to be used by itself, but it is the principal element used in forming the various alloys included under the terms *gun-metal* and *brass*, which are so extensively used in various parts of the machinery. On account of its high conductivity, copper is often used for the construction of the fire-boxes of locomotive boilers.

To insure sound and reliable castings in alloys, care

should be taken to make the proportions of the several ingredients in definite ratio to their chemical equivalents or combining weights. If this be not the case, portions of the metal will remain uncombined and the structure of the alloy will not be uniform. The proportional combining weights of the foregoing metals are, approximately, copper 63, tin 118, zinc 65, and lead 207.

Although the term *brass* is often applied indiscriminately to all alloys of copper with tin or zinc, its use, strictly speaking, should be confined to alloys of copper and zinc only: those made with copper and tin being known as gun-metal or bronze. A little zinc is usually added to the gun-metal alloys to facilitate casting.

Gun-metal or bronze.—This alloy is considerably harder than copper, and offers much greater resistance to crushing, which makes it suitable for many parts of machinery. It is easily fusible and forms good, sound and strong castings. It is, therefore, extensively used in the marine steam-engine for making cocks and valves of all descriptions, condenser and other pumps and fittings, and sometimes the condenser casings themselves. Gun-metal is much used for bearing surfaces in machinery, as it is sufficiently hard and durable to prevent excessive wear, but less so than iron or steel, so that the bearing will wear instead of the journal. The friction also between gun-metal and iron, if the surfaces be properly lubricated, is moderate and uniform, so that the bearings work smoothly. The alloys of copper and tin increase in hardness and brittleness as the percentage of tin is increased. The ordinary gun-metal used in machinery is much tougher than cast-iron, and is, therefore, more suitable than cast-iron for parts that are subject to shocks or jars. In consequence of its resistance to corrosion it is very suitable for pumps, valve-boxes, and other parts exposed to the action of water, and

is largely used for all such fittings in preference to cast-iron, though its first cost is much greater.

In ordinary practice in foundries, the metal from old castings is remelted and used over again, instead of making all new metal. To prevent deterioration of quality and insure uniformity, the old gun-metal should be first cast into ingots. This gets rid of some of the dirt and impurities, and renders the metal homogeneous in structure. The ingots are analysed to ascertain their exact composition, and the proper quantities of tin, &c., can then be added to bring the proportions of the elements in the alloy to the desired standard.

The proportions of copper, tin, and zinc in the gun-metal ordinarily used vary to some extent with the nature of the article produced. For screw-propellers and most other large engine castings, the proportions by weight are, approximately, copper 87, tin 8, and zinc 5 parts. For cocks and valves and small gun-metal castings, the composition is copper 88, tin 10, zinc 2. For engine bearings the proportion of tin may be somewhat increased, the following being an analysis of metal suitable for this purpose—copper 85, tin 10, and zinc 5. The tensile strength of good gun-metal, such as that used for bolts, &c., may be taken at about 10 to 12 tons per square inch.

Brass.—Ordinary brass which is used for cheap castings, where strength is not important, is an alloy composed of about two parts of copper to one of zinc, and is yellower in colour and much softer than gun-metal. It is not suitable for parts exposed to compression, and is not used for making engine castings. Surface condenser casings are now made of brass, containing about 70 per cent. of copper and 30 per cent. of zinc.

Brass tubes.—If the proportions of copper and zinc in the alloy be suitably arranged, the brass produced will be malleable, and may be rolled into sheets, or drawn into

tubes or wire. It is in these forms that *brass* is principally used in marine engineering; the tubes for boilers and condensers, and the internal steam and feed pipes in the boilers being generally made of this material. The proportion of copper in the boiler-tubes is generally about 68 per cent., and in the condenser-tubes 70 per cent.

Muntz-metal.—This description of brass is composed of about 60 parts of copper to 40 parts of zinc. Frequently the alloy contains one per cent. of lead to assist malleability. This material can be rolled hot into bars, plates, and sheets, and has been largely used for making rods, bolts, &c., as it possesses considerable tenacity. It has, however, been found that if Muntz-metal bolts are in contact with copper or gun-metal in sea-water, galvanic action ensues, which speedily decomposes the Muntz-metal, the zinc in the compound being destroyed.

Naval brass.—This action appears to be prevented by the addition of a little tin to the metal. An alloy composed of 62 parts of copper, 37 parts of zinc, and 1 part of tin, which for distinction is known by the name of *naval brass*, is now used instead of Muntz-metal for all such fittings. The addition of the tin, whilst apparently rendering the material impervious to galvanic action when in contact with copper or gun-metal, does not affect its strength or working qualities; and the naval brass can be forged with quite as much facility as Muntz-metal. Naval brass is also used in making the bolts for securing the planking to the iron or steel frames of composite ships, and generally for all parts in which Muntz-metal was formerly employed.

Special metal.—An alloy known as *special metal*, which, though comparatively inexpensive, possesses considerable strength, toughness, and hardness, is now used for some of the metal work on board ships. It is composed of copper 62 parts, zinc 35, tin 1, and iron 2 parts; and is some-

what similar in composition and character to the alloy known as *manganese bronze.*

Zinc.—The use of slabs of zinc for the protection of boilers was referred to in Chapter VIII. Rolled zinc is preferable to cast zinc for this purpose. Zinc is also used for coating iron that is exposed to the action of water or moisture, thus preventing its corrosion and decay. The zincing is performed by immersing the iron article, after it has been thoroughly cleaned by fire or acids, in a molten bath of zinc, by which means it becomes coated with a layer of zinc, which increases in thickness according to the length of time that the article is kept in the bath.

Lead.—Lead is not used alone in any part of the marine engine, except in the form of weights for balancing certain moving parts; and, in the earlier types of machinery, for loading the safety-valves. It is, however, used, in combination with other soft metals, for making the white-metal alloys for bearing surfaces, the compositions of which are given at page 325.

APPENDIX.

THEORETICAL INDICATOR DIAGRAMS OF COMPOUND ENGINES.

In order to determine the relative proportions of the cylinders of compound engines, and the points at which the steam should be cut off in each to produce the most uniform strains on the shafting, and enable the work due to the expansion to be most fully realised, it is necessary, in the first place, to draw theoretical indicator diagrams showing the action of the steam in the several cylinders. To simplify this operation, we will neglect the effects of the release before the end of the stroke, the compression, the resistance of the passages between the cylinders, etc. This will not materially affect the results deduced from the diagrams, and in this question simplicity is of more importance than extreme accuracy, especially as allowance can easily be made for the compression, early release, etc., after the diagrams have been drawn, if the principles involved are clearly understood.

In the following investigations:—

V has been taken to represent the volume of the large cylinder;

v the volume of the small cylinder;

U the volume of the intermediate reservoir;

R = total rate of expansion;

r = rate of expansion in high-pressure cylinder;

ρ = rate of expansion in low-pressure cylinder;

λ = ratio of cylinders; so that $R = r\lambda$;

ϕ = ratio of reservoir to h. p. cylinder; so that $U = \phi v$;

p_1 = initial absolute pressure of the steam in the high-pressure cylinder.

Then, since a volume $\frac{v}{r}$ of steam entering the high-pressure cylinder at pressure p_1, occupies finally a volume V at pressure $\frac{p_1}{R}$, we shall have $\frac{v}{r} = \frac{V}{R}$.

Engines with cranks at 0 *deg. or* 180 *deg. apart, but without an intermediate reservoir.*—In Fig. 240, OB represents the initial absolute pressure of the steam on its admission to the high-pressure cylinder, and BC is the line of pressure during admission. At C the steam is cut off and expands in the small cylinder to D, the end of the stroke, when the communication is opened to the large cylinder, and the steam exerts a forward pressure on the large piston and a back pressure on the small piston. This part of the action of the

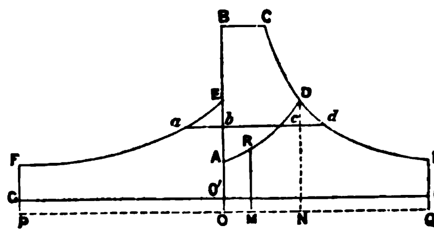


FIG. 240.

steam is represented by the two curves, D A and E F, the ordinates of D A representing the back pressures on the small piston and the corresponding ordinates of E F the forward pressures on the large piston. O P is $=$ V, the volume of the large cylinder, and O N $= v$, the volume of the small cylinder. At the end of the stroke of the large piston the communication is opened to the condenser, and the pressure falls to P G, the constant condenser pressure. These diagrams may be combined as follows:—Draw any straight line, $a\,b\,c\,d$, parallel to P O Q, and intersecting the two diagrams, and lay off on it $c\,d = a\,b$, then $b\,d = b\,c + c\,d$, represents the total volume occupied by the steam when its absolute pressure is O b, and 'd' is a point on the indicator diagram which would be formed if the steam had been expanded in the large cylinder only. By drawing a sufficient number of horizontal lines and laying off the proper distances on them, any number of points can be found, and the diagram can be reasoned about as if the whole of the action had taken place in one cylinder only. The pressure at any point in the forward stroke of the large piston, and back stroke of the small piston, is easily obtained. At any point M in the return stroke of the small piston, the total volume occupied by the steam is $(v - x) + x\frac{V}{v}$, where $x =$ N M.

Therefore the pressure M R

$$= p_1\frac{v}{r} \div \left\{v - x + x\frac{V}{v}\right\}$$

$$= p_1\frac{v}{r} \div \left\{v + x\left(\frac{R}{r} - 1\right)\right\}$$

$$= \frac{p_1\,v}{v\,r + x\,(R - r)}.$$

Engines with cranks at 0 *deg. or* 180 *deg. apart, but with an intermediate reservoir.*—This is a case that seldom occurs in practice unless the reservoir be used for the purpose of reheating the steam on its passage from the high- to the low-pressure cylinders. It is, however, interesting to examine the effect of the reservoir on the diagram, because in any case the passages between the cylinders form a sort of reservoir—in some cases not an inconsiderable one.

Let p_r = pressure in the reservoir immediately before the high-pressure cylinder exhausts into it. O A, Fig. 241, $= p_1$ = initial abso-

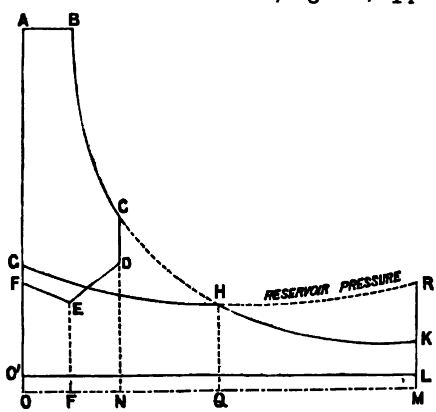


FIG. 241.

lute pressure of the steam in the high-pressure cylinder. At B the steam is cut off and expands to C, the end of the stroke of the high-pressure cylinder. At this point the communication is opened to the reservoir, and a volume, v, of steam at pressure $\frac{p_1}{r}$ is admitted to the reservoir; consequently the pressure N D will be $= \dfrac{p_r \mathrm{U} + \frac{p_1 v}{r}}{v + \mathrm{U}}$. This is, of course, equal to the initial pressure O G in the low-pressure cylinder. The steam now acts on the low-pressure piston until $\frac{1}{\rho}$th of the stroke of the low-pressure piston has been performed, when the admission to the large cylinder is cut off. At this point the steam occupies the volume $\left(1 - \frac{1}{\rho}\right) v + \mathrm{U} + \frac{\mathrm{V}}{\rho}$, and its pressure is therefore

$$= \frac{p_r \mathrm{U} + \frac{p_1}{r} v}{\left(1 - \frac{1}{\rho}\right) v + \mathrm{U} + \frac{\mathrm{V}}{\rho}} = \frac{p_r \phi + \frac{p_1}{r}}{\left(1 - \frac{1}{\rho}\right) + \phi + \frac{\lambda}{\rho}}.$$

This part of the action of the steam is represented by the curve G H in the low-pressure diagram, and D E in the high-pressure diagram. After the steam is cut off, it expands in the cylinder to the final pressure $\frac{p_1}{R}$, while in the reservoir it is compressed to the pressure p_r, represented by M R = O F. It only remains to determine p_r in order that the diagrams may be completely drawn. We can easily find p_r from the fact that a volume $\frac{V}{\rho}$ of steam at pressure $\dfrac{p_r\phi + \frac{p_1}{r}}{\left(1-\frac{1}{\rho}\right) + \phi + \frac{\lambda}{\rho}}$ occupies finally a volume V at pressure $\frac{p_1}{R}$. Therefore

$$\frac{p_r\phi + \frac{p_1}{r}}{\left(1-\frac{1}{\rho}\right) + \phi + \frac{\lambda}{\rho}} \cdot \frac{V}{\rho} = \frac{p_1}{R}V$$

$$\text{or, } p_r\,\phi + \frac{p_1}{r} = \frac{\rho\,p_1}{R}\left\{\left(1-\frac{1}{\rho}\right) + \phi + \frac{\lambda}{\rho}\right\}$$

$$\text{but } R = \lambda r$$

∴ by substitution and reduction we get

$$p_r = \frac{p_1}{R}\left\{\left(\frac{\rho-1}{\phi}\right) + \rho\right\}$$

The diagram can now be completely drawn. If the reservoir pressure p_r be not so great as the pressure of release in the high-pressure cylinder, $\frac{p_1}{r}$, there will be a fall of pressure on the admission to the reservoir, and the work due to expansion will be partly lost.

If these pressures be equal we have :—

$$\frac{p_1}{r} = \frac{p_1}{R}\left\{\frac{(\rho-1)}{\phi} + \rho\right\}$$

$$\text{or } \frac{R}{r} = \lambda = \frac{\rho-1}{\phi} + \rho$$

$$\text{or } \rho = \frac{\phi\lambda + 1}{\phi + 1}.$$

From this equation in any given case ρ can be determined so that there shall be no loss on admission to the reservoir. When $\phi = o, \rho = 1$; this is the case previously discussed. Taking $\phi = 1$, that is, taking the volume of the reservoir equal to that of the high-pressure cylinder, we have $\rho = \frac{\lambda + 1}{2}$. In this case, if λ be greater than 3, ρ will be greater than 2, consequently arrangements should be fitted to cause

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the cut-off in the low-pressure cylinder to be before half-stroke if the work due to the expansion is to be fully realised.

The following table gives a few values of rates of expansion necessary in the low-pressure cylinders of compound engines of this type when there is no fall of pressure on the admission to the reservoir.

λ =		1	2	3	4	5	6
ρ for	$\phi = 1$	1	$\frac{3}{2}$	2	$\frac{5}{2}$	3	$\frac{7}{2}$
	$\phi = 2$	1	$\frac{5}{3}$	$\frac{7}{3}$	3	$\frac{11}{3}$	$\frac{13}{3}$
	$\phi = 3$	1	$\frac{7}{4}$	$\frac{10}{4}$	$\frac{13}{4}$	4	$\frac{19}{4}$

We now pass on to consider the type of compound engine most generally used, viz., engines with two cylinders, side by side, acting on cranks at right angles to each other, and having an intermediate reservoir. The cases in which the cut-off in the low-pressure cylinder is after half-stroke, and before half-stroke respectively, must be discussed separately.

Engines with cranks at right angles to each other, having an intermediate reservoir, the cut-off in the low-pressure cylinder being after half-stroke.

It will be necessary in the first place to find an expression for the distance of the high-pressure piston from the end of its stroke, when the steam is cut off in the low-pressure cylinder.

Let O A (fig. 242) be the position of the crank of the low-pressure cylinder when the steam is cut off; O B the corresponding position of the crank of the high-pressure cylinder.

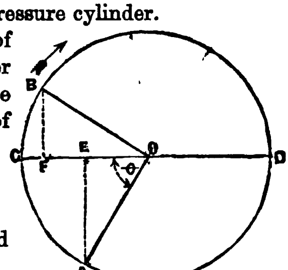


FIG. 242.

$$\text{Then } \frac{\mathrm{D\,E}}{\mathrm{D\,C}} = \frac{1}{\rho}$$

and $\frac{\mathrm{C\,F}}{\mathrm{D\,C}}$ = fraction of stroke performed by the high-pressure piston, when steam is cut off in the low-pressure cylinder.

$$= \frac{1 - \sin\theta}{2}$$

$$\frac{\mathrm{D\,E}}{\mathrm{D\,C}} = \frac{1}{\rho} = \frac{1 + \cos\theta}{2}$$

$$\text{from which, } \cos\theta = \frac{2-\rho}{\rho}$$

$$\text{and sin. } \theta = \sqrt{1-\cos^2\theta} = \frac{2}{\rho}\sqrt{\rho-1}$$

$$\text{Therefore, } \frac{\text{C F}}{\text{D C}} = \frac{\rho-2\sqrt{\rho-1}}{2\rho} = m$$

In the following investigation, for brevity's sake we will denote this by m; consequently the fraction of the small cylinder that is occupied by the steam that acts on the low-pressure piston at the point of cut-off is $=(1-m)$.

The following table gives some values of $(1-m)$ for different values of ρ:—

$\frac{1}{\rho}$ =	·5	·55	·6	·65	·7	·75	·8
1 − m =	1	·998	·990	·977	·958	·933	·9

O A (fig. 243), represents the initial pressure of steam in the high-pressure cylinder. At B the steam is cut off and expands to C, the end of the stroke of the high-pressure piston, when the com-

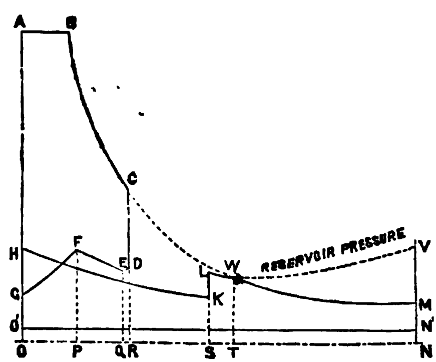


FIG. 243.

munication is opened to the reservoir and the pressure falls to R D. After this the steam expands in the reservoir and low-pressure cylinder until it is cut off in the latter. This part of the action of the steam is represented by the curve D E, Q E being the pressure at the point of cut-off. From this point the steam is compressed behind the high-pressure piston until it has completed half its return stroke, when its pressure is represented by P F. At this point the admission to the low-pressure cylinder commences, and the steam expands in

the low-pressure cylinder until the end of the return stroke of the high-pressure cylinder, when its pressure is O G.

The low-pressure diagram is easily deduced from this. The initial pressure O H is, of course, equal to the back pressure P F at the middle of the return stroke of the high-pressure piston. The steam expands in the low-pressure cylinder until half-stroke, when its pressure, S K, is obviously equal to O G. At this point the high-pressure cylinder, containing steam at the pressure R C, opens to the reservoir, and the pressure rises to S L, S L being equal to R D. From L the steam expands in the reservoir and low-pressure cylinder to W, the point of cut-off, T W being equal to Q E. From W the steam in the cylinder expands to the final pressure N M, while that in the reservoir is compressed to V, N V being equal to the initial pressure in the low-pressure cylinder. At M the communication to the condenser is opened, and the pressure falls to N N′,—the constant condenser pressure.

We will now give the algebraical expressions for the pressures at the different points, in order that the diagrams may be drawn in any given case. Since the total rate of expansion is R, the final pressure, N M, in the low-pressure cylinder is $=\frac{p_1}{R}$. The final pressure, R C, in the high-pressure cylinder is $=\frac{p_1}{r}=\frac{p_1\lambda}{R}$.

The steam in the low-pressure cylinder is expanded ρ times: consequently at the point of cut-off, the pressure, T W, is $=\frac{p_1\rho}{R}$. This is also the pressure, Q E, in the reservoir at the point of cut-off, and we have, therefore, steam at the pressure $\frac{p_1\rho}{R}$ occupying a volume $U+v\,(1-m)$. This steam is compressed behind the high-pressure piston until the beginning of the next stroke of the low-pressure piston, when its volume has been reduced to $U+\frac{v}{2}$, and its pressure has been increased to

$$\frac{p_1\rho}{R}\,\frac{U+v\,(1-m)}{U+\frac{v}{2}}=\frac{p_1\rho}{R}\,\frac{\phi+(1-m)}{\phi+\frac{1}{2}}$$

which is the initial pressure $OH=PF=NV$, in the low-pressure cylinder. This steam is driven before the high-pressure piston, and drives the low-pressure piston before it till half-stroke, when its volume is $U+\frac{V}{2}$, and the pressure S K is, therefore,

$$=\frac{p_1\rho}{R}\cdot\frac{U+v\,(1-m)}{U+\frac{V}{2}}=\frac{p_1\rho}{R}\cdot\frac{\phi+(1-m)}{\phi+\frac{1}{2}\lambda}$$

But at this point the high-pressure cylinder, containing a volume v of steam at pressure $\frac{p_1}{R}\frac{V}{v}$, opens to the reservoir, and the pressure becomes

$$=\frac{\frac{p_1\rho}{R}\left\{U+v\,(1-m)\right\}+v\frac{p_1 V}{R\,v}}{v+U+\frac{V}{2}}$$

$$=\frac{p_1}{R}\cdot\frac{\rho U+\rho\,v\,(1-m)+V}{v+U+\frac{V}{2}}$$

$$=\frac{p_1}{R}\cdot\frac{\rho\left\{\phi+(1-m)\right\}+\lambda}{1+\phi+\frac{\lambda}{2}}$$

$$=\mathrm{S\,L}=\mathrm{R\,D}.$$

Thus all the points in the diagrams have been obtained, and the diagrams can be drawn.

Fig. 243 has been drawn for cylinders having a ratio of 4 to 1: the steam being cut off at half-stroke in the small cylinder, and at ·55 of the stroke in the large cylinder. The initial pressure O A = 70 lbs., and the condenser pressure N N′ = 3 lbs. per sq. inch. The volume of the reservoir has been taken equal to the volume of the small cylinder.

It will be seen that there is a considerable drop of pressure on the admission to the reservoir, with a corresponding increase in the reservoir pressure, which produces a sudden jump in the low-pressure diagram. In an actual case this jump would be lessened by the effect of the release before the end of the stroke, and of throttling in the passages between the cylinders, and it would appear more in the form of a curve convex to O N.

It will also be seen that a large portion of the work due to expansion is lost, and consequently that the engine is not economical so far as the *theoretical* action of the steam is concerned. This sudden fall of pressure without the performance of work would possibly have the effect of heating the steam somewhat, but there would still be a loss when there is a fall of pressure, as only a percentage of this heat can be converted into mechanical work. If the work due to expansion be fully realised, this drop will become zero and we shall have—

$$R\,C = R\,D.$$

$$\text{or } \frac{p_1 V}{R\,v} = \frac{p_1}{R}\;\frac{\rho\left\{U + v\,(1-m)\right\} + V}{v + U + \frac{V}{2}}$$

$$\text{or } \frac{V}{v} = \lambda = \frac{\rho\left\{U + v(1-m)\right\} + V}{v + U + \frac{V}{2}}$$

$$\frac{\rho}{\lambda} = \frac{v + U + \frac{V}{2}}{U + v\,(1-m) + \frac{V}{\rho}}$$

But $V = \lambda\, v$, and $U = \phi\, v$. Then, by substitution, we get

$$\frac{\rho}{\lambda} = \frac{1 + \phi + \frac{\lambda}{2}}{(1-m) + \phi + \frac{\lambda}{\rho}}$$

$$\rho\left\{(1-m) + \phi\right\} + \lambda = \lambda + \lambda\left(\phi + \frac{\lambda}{2}\right).$$

Solving for λ we get,

$$\lambda = -\phi \pm \sqrt{\left\{2\,\rho\,(\overline{1-m} + \phi) + \phi^2\right.}$$

Of course the positive sign of the radical must be taken, as λ cannot be negative. From this equation, if ρ and ϕ be given, we can find the value of λ, that would prevent fall of pressure on the admission to the reservoir.

The following table gives a few values of λ, for different points of cut-off, that satisfy the foregoing conditions :—

$\frac{1}{\rho} =$	·5	·55	·6	·65	·7	·75	·8
$\rho =$	2	1·82	1·67	1·54	1·43	1·33	1·25
λ for $\phi = 1$	2	1·88	1·76	1·66	1·57	1·48	1·4
λ for $\phi = 2$	2	1·86	1·74	1·63	1·53	1·44	1·36
λ for $\phi = 3$	2	1·85	1·73	1·61	1·51	1·42	1·33

Engines with cranks at right angles to each other, having an intermediate reservoir, the cut-off in the low-pressure cylinder being before half-stroke.

The action of the steam in this case is very similar to that in the last case, with the exception that when the high-pressure cylinder opens to the reservoir the communication with the low-pressure cylinder is closed, so that the increase of pressure takes place only in the reservoir, and tends to increase the initial pressure in the low-pressure cylinder. This is shown in Fig. 244, M being the

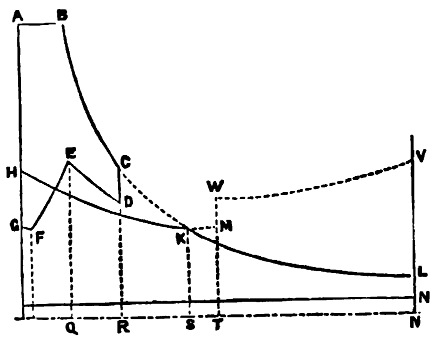
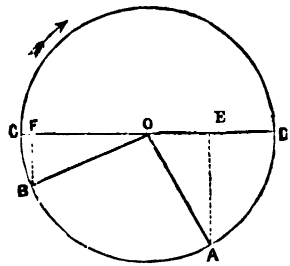


FIG. 244.

point corresponding to the release from the high-pressure cylinder. and the steam is compressed to V, NV being the initial pressure in the low-pressure cylinder.

There will be a slightly different value for the quantity $1-m$. Let O A be the position of the low-pressure crank at time of cut-off; and O B the position of the high-pressure crank.



245.

$$\frac{DF}{DC} = m = \frac{1+\sin\theta}{2}$$

$$\frac{DE}{DC} = \frac{1-\cos\theta}{2} = \frac{1}{\rho}$$

$$\therefore \cos\theta = \frac{\rho-2}{\rho};$$

and $\sin\theta = \sqrt{1-\cos^2\theta} = \frac{2}{\rho}\sqrt{\rho-1}$

$$\therefore m = \frac{1+\sin\theta}{2} = \frac{\rho+2\sqrt{\rho-1}}{2\rho}$$

$$\text{and, } 1 - m = \frac{\rho - 2\sqrt{\rho - 1}}{2\rho}.$$

The following table gives the values of $1-m$ for a few values of $\frac{1}{\rho}$.

$\frac{1}{\rho}=$	·2	·25	·3	·35	·4	·45
$1-m=$	.01	0·067	0·043	0·023	0·008	0·003

The final pressure in each of the cylinder is the same as before, viz. in the low-pressure cylinder, N L $= \frac{p_1}{R}$, and in the high pressure cylinder, R C $= \frac{p_1}{R}.\frac{V}{v} = \frac{p_1}{R}\lambda$.

The pressure at cut-off, S K $= \frac{p_1 \rho}{R}$.

At this instant the volume occupied by the steam is $U + v\ (1-m)$.

At half-stroke the volume is reduced to U, and the pressure, T M is therefore, $= \frac{p_1 \rho}{R} \frac{U + v\ (1-m)}{U} = \frac{p_1 \rho}{R}. \frac{\phi + (1-m)}{\phi}$.

At this instant the high-pressure cylinder containing a volume, v, of steam at the pressure $\frac{p_1}{R}\frac{V}{v}$ opens to the reservoir, and the pressure becomes

$$\frac{\frac{p_1 \rho}{R}\left\{U + v\ (1-m)\right\} + v\frac{p_1 V}{Rv}}{v + U}$$

$$= \frac{p_1}{R}. \frac{\rho U + \rho v\,(1-m) + V}{v + U} = \frac{p_1}{R}. \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{1 + \phi} = TW = RD$$

At the end of the stroke of the low-pressure piston this steam occupies the volume $U + \frac{v}{2}$, and its pressure is, therefore,

$$= \frac{p_1}{R}. \frac{\rho\, U + \rho\, v\ (1-m) + V}{U + \frac{v}{2}}, = \frac{p_1}{R}. \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{\phi + \frac{1}{2}}$$

which is the initial pressure, O H = N V, in the low-pressure cylinder. F is the point in the return stroke of the high-pressure piston corresponding to the point of cut-off in the low-pressure cylinder. Conse-

quently the pressure P F = S K = $\frac{p_1 \rho}{R}$. The pressure O G is = T M. Thus all the points have been determined and the diagrams can be drawn in any given case.

Fig. 244 has been drawn for the same engine as Fig. 243 the total expansion was also the same, the only difference being that in this case the steam has been cut off at ·4 of the stroke in the low-pressure cylinder instead of ·55 as in the previous example.

There is in this case also a fall of pressure on the admission to the reservoir, with a corresponding increase in the pressure. The drop is, however, not so great nor so injurious, as it tends to increase the initial pressure of the steam in the low-pressure cylinder instead of the pressure at the middle of the stroke. All the work due to the expansion, however, is not realised, and there is still a considerable loss. If there were no drop we should have R C = R D,

$$\text{or } \frac{p_1}{R} \cdot \frac{V}{v} = \frac{p_1}{R} \frac{\rho \left\{ U + v\,(1-m) \right\} + V}{v + U}$$

Putting V = λ v, and U = ϕ v, we get,

$$\frac{\lambda}{\rho} = \frac{(1-m) + \phi + \frac{\lambda}{\rho}}{1+\phi}$$

$$\lambda\,(1+\phi) = \rho \left\{ (1-m) + \phi \right\} + \lambda$$

$$\frac{\lambda}{\rho} = \frac{(1-m) + \phi}{\phi}.$$

From this we get the following table:—

$\frac{1}{\rho}$ =		·2	·25	·3	·35	·4	·45
ρ =		5·0	4·0	3·33	2·86	2·50	2·222
λ for	$\phi = 1$	5·5	4·27	3·48	2·92	2·52	2·228
	$\phi = 2$	5·25	4·13	3·41	2·89	2·51	2·225
	$\phi = 3$	5·17	4·09	3·38	2·88	2·507	2·224

The numerical values of the expressions for the pressures at the different points can be easily found in any given case, as the values of $(1-m)$ are given in the tables. To facilitate calculation the following summary of the preceding investigations is given.

In the case in which the cut-off in the low-pressure cylinder is after half-stroke, we have,

$$O\,A = p_1,\ N\,M = \frac{p_1}{R}$$

$$R\,C = \frac{p_1}{R}\lambda = N\,M \times \lambda$$

$$R\,D = S\,L = \frac{p_1}{R}.\ \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{1 + \phi + \frac{\lambda}{2}}$$

$$= N\,M \times \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{1 + \phi + \frac{\lambda}{2}}$$

$$Q\,E = T\,W = \frac{p_1}{R}\rho = N\,M \times \rho$$

$$P\,F = O\,H = \frac{p_1\,\rho}{R}.\ \frac{\phi + (1-m)}{\phi + \frac{1}{2}}$$

$$= Q\,E \times \frac{\phi + (1-m)}{\phi + \frac{1}{2}}$$

$$O\,G = S\,K = \frac{p_1\,\rho}{R}.\ \frac{\phi + (1-m)}{\phi + \frac{\lambda}{2}}$$

$$= Q\,E \times \frac{\phi + (1-m)}{\phi + \frac{\lambda}{2}}$$

When the cut-off in the low-pressure cylinder takes place before half-stroke, we have

$$O\,A = p_1;\ NL = \frac{p_1}{R};\ R\,C = \frac{p_1}{R}\lambda = N\,L \times \lambda$$

$$R\,D = \frac{p_1}{R}\ \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{1 + \phi}$$

$$= N\,L \times \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{1 + \phi}$$

$$Q\,E = O\,H = \frac{p_1}{R}\ \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{\phi + \frac{1}{2}}$$

$$= N\,L \times \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{\phi + \frac{1}{2}}$$

$$= \text{also, } R\,D \times \frac{\phi + 1}{\phi + \frac{1}{2}}$$

$$P\,F = S\,K = \frac{p_1}{R}\rho = N\,L \times \rho$$

$$O\,G = \frac{p_1 \rho}{R}\,\frac{\phi + (1-m)}{\phi}$$

$$= S\,K \times \frac{\phi + (1-m)}{\phi}.$$

Thus all the points in the diagram can be readily determined when the initial pressure, rates of expansion, and the ratios of the low-pressure cylinder and reservoir to the high-pressure are given.

On reference to the tables on pages 629 and 632, it will be seen that with this type of engine, if the ratio of cylinders is more than two to one, the cut-off in the low-pressure cylinder should be arranged to take place before half-stroke, to prevent loss from sudden expansion, and consequently an expansion valve would be required on the low-pressure cylinder in order to enable the full benefit of the expansion of the steam to be realised.

Three-cylinder Compound Engines.—The investigation of the action of the steam in three-cylinder compound engines with cranks at equal angles to each other must be divided into three parts.

1. When both low-pressure cylinders are open to the reservoir at the time that the high-pressure cylinder exhausts into it; that is, when the cut-off in each low-pressure cylinder is after 0·75 of the stroke.

2. When only one low-pressure cylinder is open to the reservoir when the high-pressure exhausts into it; that is, when the cut-off in each low-pressure cylinder is between 0·25 and 0·75 of the stroke.

3. When neither of the low-pressure cylinders is open to the reservoir when the high-pressure cylinder exhausts into it; that is, when the cut-off in each low-pressure cylinder is before 0·25 of the stroke.

(1) *Cut-off in the low-pressure cylinder after* 0·75 *of the stroke.*

By reference to Fig. 246, showing the positions of the several cranks, when the steam is exhausted from the high-pressure cylinder to the reservoir, it will be seen that the action of the steam in this case would be most disadvantageous. If O P and O Q represent the low-pressure cranks and O R the high-pressure crank, it is clear that while one low-pressure cylinder gets steam at one-quarter stroke the other does not get its supply until the piston has traversed three-fourths of its stroke and is near the point at which the admission of steam is cut off by the slide-valve. The work done by the two low-pressure cylinders would in consequence be

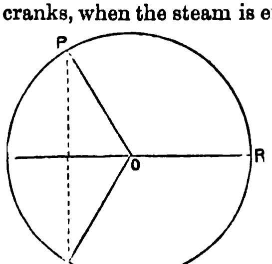


FIG. 246.

very unequal, one of the cylinders being practically of very little use. No further discussion of this case is necessary, as it would not occur in any well-designed engine, as it is evident from the foregoing that in all three-cylinder compound engines the cut-off in the low-pressure cylinders should be arranged to take place before 0·75 of the stroke.

(2) *Cut-off in the low-pressure cylinders between* 0·25 *and* 0·75 *of the stroke.*

This is the most general case that occurs in practice.

Let v = volume of the high-pressure cylinder.

V = " each low "

U = " the intermediate reservoir.

R = total ratio of expansion.

r = ratio of expansion in the high-pressure cylinder.

ρ = " " each low "

λ = ratio of each low-pressure cylinder to the high-pressure cylinder.

ϕ = ratio of the intermediate reservoir to the high-pressure cylinder.

So that
$$V = \lambda v$$
$$U = \phi v$$
$$R = 2 \lambda r.$$

Also, for brevity, let the symbols a and β represent the low-pressure cylinders, and γ the high-pressure cylinder.

In the first place it will be necessary to investigate expressions for the distances of the high-pressure piston from the ends of the stroke when each of the low-pressure pistons are at the points of cut-off.

In Fig. 247, let O P, O Q represent the positions of the cranks of the low-pressure cylinders a and β respectively, and O R the position of the crank of the high-pressure cylinder γ at the point of cut-off in a.

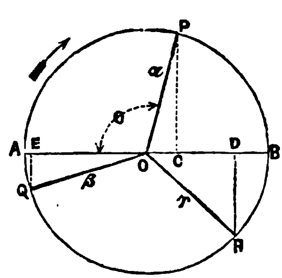


Fig. 247.

Then $\frac{A\,C}{A\,B} = \frac{1}{\rho} = \frac{1 - \cos.\,\theta}{2}$

or $\cos.\,\theta = \frac{\rho - 2}{\rho}$

$\text{Sin.}\,\theta = \sqrt{1 - \cos^2\theta} = \frac{2}{\rho}\sqrt{\rho - 1}$

At this point the distance of the high-pressure piston from the end of its stroke is represented by B D, so that the fraction of the high-pressure cylinder open to the reservoir is represented by,

$$\frac{\text{A D}}{\text{A B}} = \frac{1+\cos(\theta-60^\circ)}{2}$$

$$= \frac{1+\frac{1}{2}\cos\theta+\frac{1}{2}\sqrt{3}\,.\sin\theta}{2}$$

$$= \frac{3\rho+2\sqrt{3(\rho-1)}-2}{4\rho}$$

For brevity's sake call this l.

$$\frac{\text{A E}}{\text{A B}} = \frac{1+\cos(\theta+60^\circ)}{2}$$

$$= \frac{1+\frac{1}{2}\cos\theta-\frac{1}{2}\sqrt{3}\sin\theta}{2}$$

$$= \frac{3\rho-2\sqrt{3(\rho-1)}-2}{4\rho}$$

Let this expression be denoted by m.

The following table gives values of l and m for different values of ρ :—

$\frac{1}{\rho}=$	0·25	0·30	0·35	0·40	0·45	0·50	0·55	0·60	0·65	0·70	0·75
$\rho=$	4·0	3·33	2·86	2·50	2·22	2·0	1·82	1·67	1·54	1·43	1·33
$l=$	1·0	.997	·988	·974	·955	·933	·906	·875	·838	·797	·748
$1-l=$	0·0	·003	·012	·026	·045	·067	·094	·125	·162	·203	·252
$m=$	·250	·203	·162	·126	·094	·067	·044	·026	·010	·003	0·00

This table will be useful in facilitating calculation of the diagrams.

In investigating the nature of the diagram let us first assume that the cut-off and the final pressures in each of the low-pressure cylinders are the same.

In this case the final pressure in each of the low-pressure cylinders is $=\frac{p_1}{R}$; and the pressure at cut-off in each of the low-pressure cylinders $=\frac{\rho p_1}{R}$.

Let the theoretical diagrams be represented by figs. 248, 249, and 250: fig. 248 being from the high-pressure cylinder γ and figs. 249 and 250 from the low-pressure cylinders α and β respectively. Also let the pressures at the different points be denoted by the letter p, with the suffix corresponding to the number of the point on the diagrams.

First trace the action of the steam in the cylinder α.

The total volume occupied by the steam at the point of cut-off in a is, $= U + \frac{V}{\rho} + lv$

and the pressure at this point is $= \frac{\rho p_1}{R} = p_{12}$

This is also $= p_{16}$ and to p_5.

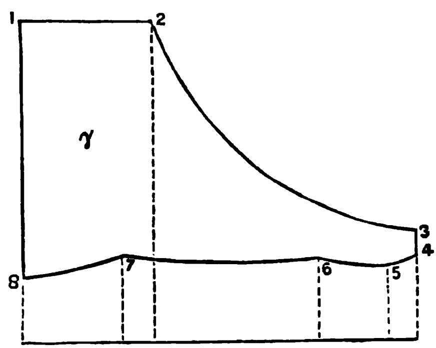


FIG. 248.

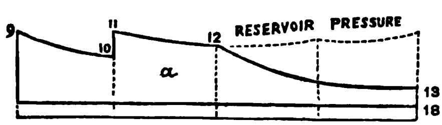


FIG. 249.

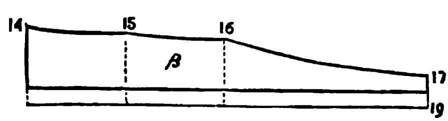


FIG. 250.

At quarter-stroke of a, *just after* the high-pressure cylinder has exhausted into the reservoir, the volume occupied by the steam is,

$$= U + v + \frac{V}{4}$$

and the pressure is therefore

$$= \frac{\rho p_1}{R} \cdot \frac{U + \frac{V}{\rho} + lv}{U + \frac{V}{4} + v}$$

$$= \frac{\rho p_1}{R} \cdot \frac{\phi + \frac{\lambda}{\rho} + l}{\phi + \frac{\lambda}{4} + 1}$$

$$= p_4 = p_{11}.$$

The volume occupied by the steam in the reservoir, &c., *immediately before* the high-pressure cylinder exhausts is $= U + \frac{V}{4}$; and its pressure is represented by p_{10}.

At this point a volume v, of steam at a pressure $\frac{p_1}{r}$ is admitted to the reservoir from the high-pressure cylinder and the pressure rises to p_{11}, the steam then occupying the volume,

$$\left(U + \frac{V}{4} + v\right)$$

Therefore we have,

$$p_{10}\left(U + \frac{V}{4}\right) + \frac{p_1}{r} v = p_{11}\left(U + \frac{V}{4} + v\right) = \frac{\rho p_1}{R}\left(U + \frac{V}{\rho} + lv\right)$$

$$\text{or } p_{10} = \frac{\frac{\rho p_1}{R}\left(U + \frac{V}{\rho} + lv\right) - \frac{p_1}{r} v}{U + \frac{V}{4}}$$

$$\text{But } r = \frac{R}{2\lambda}$$

$$\text{Therefore } \frac{p_1}{r} = \frac{2\lambda p_1}{R}$$

$$\text{And } p_{10} = \frac{p_1}{R} \frac{\rho\left(U + lv\right) + V - 2\lambda v}{U + \frac{V}{4}}$$

$$= \frac{p_1}{R} \frac{\rho\left(\phi + l\right) - \lambda}{\phi + \frac{\lambda}{4}}$$

This is also $= p_8$.

After the steam is cut off in a, the steam remaining in the reservoir is compressed behind the high-pressure piston, until the commencement of the stroke of β.

The volume of steam in the reservoir at cut-off of a is $= U + lv$.

At the commencement of the stroke of β this has been altered to $U + \frac{8}{4} v$; and the pressure is, therefore,

$$= \frac{\rho p_1}{R} \cdot \frac{U + lv}{U + \frac{3}{4}v}$$

$$= \frac{\rho p_1}{R} \cdot \frac{\phi + l}{\phi + \cdot 75} = \text{initial in } \beta = p_{14}$$

$$= \text{also } p_6.$$

This steam acts on the piston of β and is acted on by the high-pressure piston till quarter-stroke of β, when a is ready to commence its return stroke. The volume occupied by the steam at this point is $= U + \frac{1}{4}\left(V + v\right)$, and the pressure is, therefore,

$$= \frac{\rho p_1}{R} \cdot \frac{U + l v}{U + \frac{1}{4}(V + v)}$$

$$= \frac{\rho p_1}{R} \cdot \frac{\phi + l}{\phi + \frac{1}{4}(\lambda + 1)} = \text{initial pressure in } a, = p_9 = p_{15} = p_7$$

Until the point of cut-off in β, this steam acts on the two low-pressure pistons, and is acted on by the high-pressure piston. At the point of cut-off in β, the volume occupied by the steam is

$$= U + \frac{V}{\rho} + (1 - l)V + m v$$

and the pressure is, therefore,

$$= \frac{\rho p_1}{R} \cdot \frac{U + l v}{U + \frac{V}{\rho} + \left(1 - l\right)V + m v}$$

$$= \frac{\rho p_1}{R} \cdot \frac{\phi + l}{\phi + \lambda\left(\frac{1}{\rho} + 1 - l\right) + m}$$

$= p_{16}$, the pressure at the point of cut-off in β.

By the assumption previously made, viz.:—that the pressures at the points of cut-off in the cylinders a and β are to be the same, this must be equal to $\frac{\rho p_1}{R}$.

Therefore,

$$\frac{\phi + l}{\phi + \lambda\left(\frac{1}{\rho} + 1 - l\right) + m} \text{ must be} = 1$$

$$\text{or } \phi + \lambda\left(\frac{1}{\rho} + 1 - l\right) + m = \phi +$$

$$\lambda = \frac{l - m}{\frac{1}{\rho} + 1 - l}$$

From this we see that only when a certain relation exists between the ratios of cylinders and the point of cut-off in the low-pressure cylinders, can the final pressures and rates of expansion be the same in each of the low-pressure cylinders.

From the foregoing equation the necessary ratios of cylinders for certain points of cut-off, in order to make the final pressures in the low-pressure cylinders the same, can be readily found, and a few values are given below:—

$\frac{1}{\rho}=$	0·25	0·30	0·35	0·40	0·45	0·50	0·55	0·60	0·65	0·70	0·75
$l=$	4·0	3·33	2·86	2·50	2·22	2·0	1·82	1·67	1·54	1·43	1·33
$\lambda=$	3·0	2·62	2·28	1·99	1·75	1·52	1·34	1·17	1·02	0·88	0·75

From this table we may see that in the majority of cases in general practice, the expansion and cut-off will not be exactly the same in the two low-pressure cylinders. In general, if the steam is cut off at the same part of the stroke in each of the two low-pressure cylinders, the final pressures will be different; or if the final pressures be the same the points of cut-off will be different.

First assume the final pressures the same in each of the low-pressure cylinders, but the cut-offs different.

Let ρ represent the ratio of expansion in a.

" ρ_1 " " β.

The same method of reasoning must be applied as in the previous case, and it will be found that the only pressure that is altered is p_{16}.

This becomes

$$p_{16}=\frac{\rho p_1}{\text{R}}\cdot\frac{\phi+l}{\phi+\lambda\left(\frac{1}{\rho_1}+1-l_1\right)+m_1}$$

Where l_1 and m_1 are the values of l and m corresponding to the ratio of expansion ρ_1,

The final pressure in β, p_{17}, is, therefore,

$$=\frac{\rho}{\rho_1}\frac{p_1}{\text{R}}\cdot\frac{\phi+l}{\phi+\lambda\left(\frac{1}{\rho_1}+1-l_1\right)+m_1}.$$

But by supposition this is equal to the final pressure in a,

$$\text{or }\frac{p_1}{\text{R}}=\frac{\rho}{\rho_1}\cdot\frac{p_1}{\text{R}}\,\frac{\phi+l}{\phi+\lambda\left(\frac{1}{\rho}+1-l_1\right)+m_1}$$

$$\text{or, } \frac{\rho}{\rho_1} = \frac{\phi + \lambda\left(\frac{1}{\rho_1} + 1 - l_1\right) + m_1}{\phi + l}$$

$$\rho\,(\phi + l) = \rho_1 \left\{ \phi + \lambda\,(1 - l_1) + m_1 \right\} + \lambda$$

$$\text{or, } \rho_1 = \frac{\rho\,(\phi + l) - \lambda}{\lambda\,(1 - l_1) + \phi + m_1}.$$

As ρ and ρ_1 are not very different, it will be sufficiently accurate for practical purposes to take l_1 and m_1 the same as l and m. In this case the value of ρ_1 for any given value of ρ can be obtained.

Secondly.—Assume that the cut-off is the same in each of the low-pressure cylinders, so that, generally, the final pressures would be different.

Let p_a represent the final pressure in a.
" p_β " " β.

We shall be sufficiently accurate if we assume that $\frac{p_1}{R}$, which is the pressure due to the total expansion, is a mean between p_a and p_β.

$$\text{or, } \tfrac{1}{2}\left(p_a + p_\beta\right) = \frac{p_1}{R}$$

The expressions for the pressures at each of the points will be exactly similar to those previously given, with the exception that p_a must be substituted for $\frac{p_1}{R}$.

The expression for the pressure at the point of cut-off in β, is

$$p_{16} = \rho\, p_a \frac{\phi + l}{\phi + \lambda\left(\frac{1}{\rho} + 1 - l\right) + m}$$

and this must by hypothesis be $= \rho\, p_\beta$.

$$\text{Therefore } \frac{p_a}{p_\beta} = \frac{\phi + \lambda\left(\frac{1}{\rho} + 1 - l\right) + m}{\phi - l}$$

From this equation the ratios of the final pressures in the two low-pressure cylinders to each other can be found; and then from the equation,

$$\tfrac{1}{2}\left(p_a + p_\beta\right) = \frac{p_1}{R}$$

the actual final pressures may be obtained.

In the following Table are given, for a few ratios of cylinders:—

1. The cut-off, $\frac{1}{\rho_1}$, in β, for certain values of $\frac{1}{\rho}$, the cut-off in a, when the final pressures are supposed to be the same; and,

2. The ratio of the final pressures in the two low-pressure cylinders, when the steam is cut off at $\frac{1}{\rho}$th the stroke in each; or $\frac{p_a}{p_\beta} = \delta$ suppose.

$\lambda =$	1			2			3		
$\phi =$	1	2	3	1	2	3	1	2	3
$\frac{1}{\rho} =$	0·3	0·3	0·3	0·3	0·3	0·3	0·3	0·3	0·3
$\frac{1}{\rho_1} =$	0·226	0·252	0·264	0·249	0·264	0·275	0·270	0·282	0·285
$\delta =$	0·755	0·840	0·880	0·830	0·880	0·915	0·9	0·94	0·95
$\frac{1}{\rho} =$	0·4	0·4	0·4	0·4	0·4	0·4	0·4	0·4	0·4
$\frac{1}{\rho_1} =$	0·312	0·340	0·360	0·356	0·368	0·376	0·398	0·399	0·4
$\delta =$	0·78	0·85	0·90	0·89	0·92	0·94	0·995	0·997	1·0
$\frac{1}{\rho} =$	0·5	0·5	0·5	0·5	0·5	0·5	0·5	0·5	0·5
$\frac{1}{\rho_1} =$	0·4	0·45	0·46	0·495	0·5	0·5	0·57	0·545	0·53
$\delta =$	0·8	0·9	0·92	0·99	1·0	1·0	1·14	1·09	1·06
$\frac{1}{\rho} =$	0·6	0·6	0·6	0·6	0·6	0·6	0·6	0·6	0·6
$\frac{1}{\rho_1} =$	0·558	0·576	0·582	0·678	0·648	0·636	0·78	0·72	0·69
$\delta =$	0·93	0·96	0·97	1·13	1·08	1·06	1·3	1·2	1·15
$\frac{1}{\rho} =$	0·7	0·7	0·7	0·7	0·7	0·7	0·7	0·7	0·7
$\frac{1}{\rho_1} =$	0·742	0·728	0·721	0·91	0·84	0·798	1·092	0·952	0·882
$\delta =$	1·06	1·04	1·03	1·3	1·2	1·14	1·56	1·36	1·26

To facilitate calculation we will summarise the formulæ for the pressures at the different points; assuming the cut-off and final pressure to be the same in each of the low-pressure cylinders.

$p_1 = p_2 =$ initial pressure in the cylinders.

$$p_3 = \frac{p_1}{r} = 2\,\frac{p_1\lambda}{R}.$$

$$p_4 = p_{11} = \frac{\rho p_1}{R}.\ \frac{\phi + \dfrac{\lambda}{\rho} + l}{\phi + \dfrac{\lambda}{4} + 1}$$

$$p_5 = p_{12} = p_{16} = \frac{\rho p_1}{R}.$$

$$p_6 = p_{14} = \frac{\rho p_1}{R}.\ \frac{\phi + l}{\phi + 0{\cdot}75}$$

$$p_7 = p_9 = p_{15} = \frac{\rho p_1}{R}.\ \frac{\phi + l}{\phi + \frac{1}{4}(\lambda + 1)}$$

$$p_8 = p_{10} = \frac{p_1}{R}.\ \frac{\rho(\phi + l) - \lambda}{\phi + \dfrac{\lambda}{4}}$$

$$p_{13} = p_{17} = \frac{p_1}{R}.$$

$p_{18} = p_{19} =$ condenser pressure $=$ 3 to 4lbs. absolute.

In cases in which the points of cut-off, or the final pressures, are different in the two low-pressure cylinders, the necessary modifications must be made in the above formulæ as explained on page 640.

(3) *Cut-off in low-pressure cylinders before quarter-stroke.*

In this case the receiver is never in communication with more than one low-pressure cylinder at the same time.

This case would seldom occur in practice except when the engines were being worked at reduced power, for which the expansion valves on the low-pressure cylinders, if they are fitted, should be set at an early point of cut-off. In engines in which the combined volume of the low-pressure cylinders is more than four times that of the high-pressure cylinder, loss from sudden expansion on admission to the receiver can only be avoided by cutting off the admission to the low-pressure cylinders before quarter stroke.

We will as before, in the first place, find expressions for the distance of the high-pressure piston from the end of its stroke when the low-pressure pistons are at the points of cut-off.

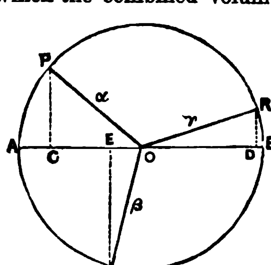


Fig. 251.

In Fig. 251, O P and O Q represent the low-pressure cranks,

O P being in the position at which steam is cut off in *a*. O R is the corresponding position of the high-pressure crank.

$$\frac{AC}{AB} = \frac{1}{\rho} = \frac{1-\cos\theta}{2}$$

$$\text{or } \cos\theta = \frac{\rho-2}{\rho}.$$

$$\text{and } \sin\theta = \frac{2}{\rho}\sqrt{\rho-1}$$

$$\frac{BD}{AB} = \frac{1+\cos(120^\circ+\theta)}{2}$$

$$= \frac{1-\frac{1}{2}\cos\theta-\frac{1}{2}\sqrt{3}\sin\theta}{2}$$

$$= \frac{1-\frac{\rho-2}{2\rho}-\frac{2\sqrt{3}\sqrt{\rho-1}}{2\rho}}{2}$$

$$= \frac{\rho+2-2\sqrt{3(\rho-1)}}{4\rho}$$

Call this l for convenience as before.

$$\frac{AE}{AB} = \frac{1-\cos(120-\theta)}{2}$$

$$= \frac{1+\frac{1}{2}\cos\theta-\frac{1}{2}\sqrt{3}\sin\theta}{2}$$

$$= \frac{1+\frac{\rho-2}{2\rho}-\frac{2\sqrt{3}\sqrt{\rho-1}}{2\rho}}{2}$$

$$= \frac{3\rho-2-2\sqrt{3(\rho-1)}}{4\rho}$$

Let this be denoted by m.

To facilitate calculation the values of l and m for a few points of cut-off are given in the following table :—

$\frac{1}{\rho} =$	0·1	0·125	0·15	0·175	0·2
$\rho =$	10	8	6·67	5·71	5
$l =$	0·04	0·026	0·016	0·008	0·003
$m =$	0·44	0·4	0·365	0·33	0·303

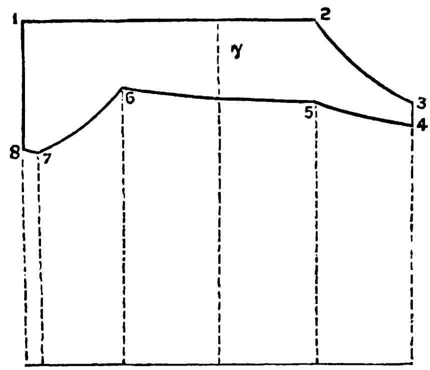


FIG. 252.

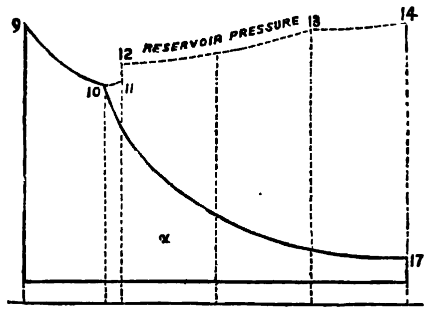


FIG. 253.

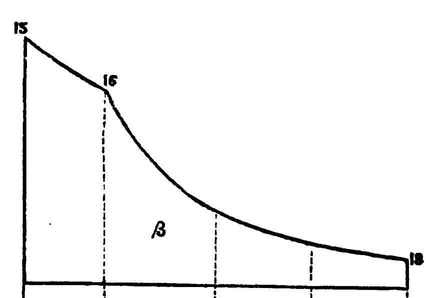


FIG. 254.

First, assume the cut-off and final pressure to be the same in the two low-pressure cylinders. The theoretical diagrams are shown in Figs. 252, 253, and 254. The final pressure in each of the low-pressure cylinders is $=\frac{p_1}{R}$ and at the point of cut-off $=\frac{\rho p_1}{R}$.

The volume occupied by the steam at the instant of cut-off is

$$= U + \frac{V}{\rho} + l v$$

At the commencement of the stroke of a, the volume occupied by the steam was $= U + \frac{1}{4} v$, and its pressure at that time, therefore,

$$\text{was} = \frac{\rho p_1}{R} \cdot \frac{U + \frac{V}{\rho} + l v}{U + \frac{1}{4} v}$$

$$= \frac{\rho p_1}{R} \cdot \frac{\phi + \frac{\lambda}{\rho} + l}{\phi + \frac{1}{4}}$$

$$= \text{initial pressure in } a = p_9.$$

At the instant of cut-off in a, the volume of steam in the reservoir is $= U + l v$. This is compressed behind the high-pressure piston until the end of its stroke, when the volume it occupies is $= U$, and the pressure is increased to

$$\frac{\rho p_1}{R} \cdot \frac{U + l v}{U} = \frac{\rho p_1}{R} \cdot \frac{\phi + l}{\phi}.$$

This is $= p_8$, the back pressure on the high-pressure piston at the end of the return stroke, and equal also to p_{11}. At this point the high-pressure cylinder containing a volume v of steam at a pressure $\frac{p_1}{r}$ exhausts into the reservoir, and the pressure falls to p_4.

Therefore

$$p_8 U + \frac{p_1}{r} v = p_4 (U + v)$$

$$\text{or } p_4 = \frac{p_8 U + \frac{p_1}{r} v}{U + v}$$

$$\text{But } r = \frac{R}{2\lambda}$$

$$\therefore p_4 = \frac{\frac{\rho p_1}{R} (U + l v) + \frac{2 \lambda p_1}{R} v}{U + v}$$

$$= \frac{p_1}{R} \cdot \frac{\rho (\phi + l) + 2 \lambda}{\phi + 1}$$

= pressure in receiver after the high-pressure cylinder has ex-

hausted into it, p_{12}. This is compressed behind the high-pressure piston for one-quarter of its stroke, when the low-pressure piston of β, is about to commence its stroke, and the initial pressure in β, p_{15}, will, therefore, be

$$= \frac{p_1}{R} \cdot \frac{\rho (U + lv) + 2 \lambda v}{U + \frac{3}{4} v}$$

$$= \frac{p_1}{R} \cdot \frac{\rho (\phi + l) + 2\lambda}{\phi + 0{\cdot}75}$$

This also $= p_5 = p_{13}$.

At $\frac{1}{\rho}$th of the stroke of β, this steam occupies a volume

$$= U + \frac{V}{\rho} + m v$$

and its pressure is therefore,

$$= \frac{p_1}{R} \cdot \frac{\rho (U + l v) + 2 \lambda v}{U + \frac{V}{\rho} + m v}$$

$$= \frac{p_1}{R} \cdot \frac{\rho (\phi + l) + 2\lambda}{\phi + \frac{\lambda}{\rho} + m}$$

which is the pressure at the point of cut-off in β, and by hypothesis should be,

$$= \frac{\rho p_1}{R}$$

$$\text{or } \rho = \frac{\rho (\phi + l) + 2\lambda}{\phi + \frac{\lambda}{\rho} + m}$$

$$\rho (\phi + \frac{\lambda}{\rho} + m) = \rho (\phi + l) + 2 \lambda$$

$$\text{or } \rho = \frac{\lambda}{m - l}$$

$$\text{or } \lambda = \rho (m - l).$$

From this equation the ratio between the proportions of the cylinders and the points of cut-off in the low-pressure cylinders necessary to render the cut-off and final pressure the same in each of the low-pressure cylinders may be easily found. The following are a few values which satisfy the foregoing conditions:—

$\frac{1}{\rho} =$	0·1	0·125	0·15	0·175	0·2
$\rho =$	10·0	8·0	6·67	5·71	5·0
$\lambda =$	4·0	2·99	2·327	1·85	1·5

We see, therefore, as in the previous case, that generally there will be some difference either in the points of cut-off, or in the final pressures in the two low-pressure cylinders.

First assume the final pressures to be the same in each.

Let ρ be the ratio of expansion in the cylinder a, and ρ_1 the ratio of expansion in β. Then the formulæ will be precisely the same as before, except that ρ_1, l_1, and m_1, must be substituted for ρ, l and m, in all the expressions that involve the cut-off in β.

The pressure at point of cut-off in β

$$p_{16} = \frac{p_1}{R} \cdot \frac{\rho(\phi + l) + 2\lambda}{\phi + \frac{\lambda}{\rho_1} + m_1}$$

Therefore the final pressure in β will be

$$= \frac{1}{\rho_1} \cdot \frac{p_1}{R} \cdot \frac{\rho(\phi + l) + 2\lambda}{\phi + \frac{\lambda}{\rho_1} + m_1}$$

This by hypothesis is $= \frac{p_1}{R}$,

$$\text{or } \frac{1}{\rho_1} \cdot \frac{\rho(\phi + l) + 2\lambda}{\phi + \frac{\lambda}{\rho_1} + m_1} = 1$$

$$\text{or } \rho_1(\phi + m_1) + \lambda = \rho(\phi + l) + 2\lambda$$

$$\rho_1 = \frac{\rho(\phi + l) + \lambda}{\phi + m_1}$$

Next, suppose the point of cut-off in the two cylinders to be the same, in which case the final pressures will generally be different. Let these pressures be denoted by p_a and p_β, as in the former case. and suppose $\frac{1}{2}(p_a + p_\beta) = \frac{p_1}{R}$.

The formulæ for the pressures at the various points will be the same as before, except that p_a must be substituted for $\frac{p_1}{R}$.

The pressure at the point of cut-off in β will then be

$$p_a \cdot \frac{\rho(\phi + l) + 2\lambda}{\phi + \frac{\lambda}{\rho} + m}$$

This by hypothesis is $= \rho p_\beta$

$$\therefore \frac{p_a}{p_\beta} = \frac{\rho(\phi + m) + \lambda}{\rho(\phi + l) + 2\lambda}$$

From this and the equation $\frac{1}{2}(p_a + p_\beta) = \frac{p_1}{R}$, the actual values of p_a and p_β in any given instance may be readily found.

GEOMETRICAL REPRESENTATION OF THE TWISTING MOMENTS ON THE CRANK-SHAFT.

For the sake of simplicity, suppose the obliquity of the connecting-rod and the weights of the reciprocating parts to be neglected.

Let Fig. 255 represent the crank-circle, O being the centre of the shaft; and suppose the forward pressure to be constant throughout the stroke, and equal to P. Then, it is clear that when the crank is in any position O C, making an angle θ with the line of dead points, the twisting moment exerted will be $= P \times L \sin \theta$; where L represents the length of the crank. If, therefore, the crank-circle be divided into any number of equal parts, each subtending an angle a, the successive twisting moments will be, $P\,L \sin a$, $P\,L \sin 2a$, $P\,L \sin 3a$, &c.

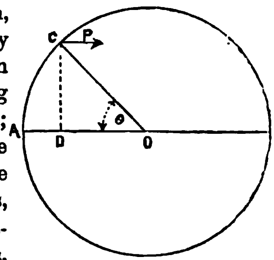


FIG. 255.

The base line of the diagram of twisting moments is taken to represent the circumference of the crank-circle, and is divided into a number of equal parts representing equal angles of the crank with the line of dead points. Ordinates are set up at these divisions equal to the twisting moment ($P \times L \sin \theta$), for the corresponding angle, and a fair curve is drawn through the ends of the ordinates thus obtained.

In the case under consideration, the curve will be symmetrical,

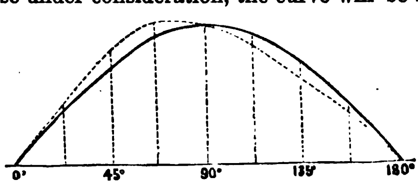


FIG. 256.

the obliquity of the connecting-rod having been neglected, so that the piston is supposed to have exact harmonic motion. This *curve of sines* is shown by the *full lines* in Fig. 256.

When the obliquity of the connecting-rod is taken into account the difference of speed of piston at the opposite ends of the stroke will be found to destroy the symmetry of the curve of twisting moments. By reference to Fig. 257, it will be seen that, if ϕ be the angle of the connecting-rod when the crank makes an angle θ with the line of dead points, the twisting moment, instead of being $= P \times L \sin \theta$ simply, is

$$= Q \times OC \sin OCD = Q \times L \sin(\theta + \phi)$$

where Q = thrust on the connecting rod $= \dfrac{P}{\cos \phi}$.

$\therefore$ the twisting moment in this case is

$$= P \times L \frac{\sin(\theta + \phi)}{\cos \phi}$$

$$= P.L (\sin \theta + \cos \theta \tan \phi)$$

The second term in the bracket is always small, and when θ is greater than 90°, will be negative: so that it is clear that the curve

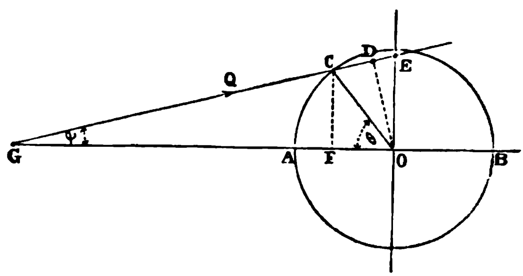


FIG. 257.

will be fuller in the first quarter and less in the second quarter revolution, as shown by the dotted lines in Fig. 257.

If the crank-circle be divided into 16 equal parts, each subtending an angle of 22½°, the successive multipliers of P.L. will be

	1	2	3	4	5	6	7	8	9
Connecting-rod, infinite	0	0·383	0·707	0·924	1·0	0·924	0·707	0·383	0
Connecting-rod = four cranks	0	0·428	0·834	1·015	1·0	0·833	0·580	0·338	0

In practice it is generally convenient to take P in tons, and L in inches, so that the resulting twisting moments are obtained in inch-tons.

We now proceed to explain how actual indicator diagrams taken from the cylinders of an engine, may be utilised to show the variations of the twisting moments on the crank-shaft throughout the stroke. The pressure on the piston will in this case vary at each point of the stroke, instead of being constant as before assumed, and

the curve of twisting moments will be much less regular, and will fall very rapidly from the point at which expansion begins.

The crank-shaft is supposed to revolve uniformly as before, so that the base line of the diagram of twisting moments is still divided into equal parts, to represent the successive angles of the crank. Ordinates are drawn across the indicator diagram at the points corresponding to the respective positions of the piston for the several angles of the crank, which will give the pressure of steam per square inch on the piston for the given angle of the crank. This pressure multiplied by the area of the piston, by the length of the crank, and by the quantity represented by $\frac{\sin(\theta+\phi)}{\cos\phi}$, will give the corresponding twisting moment on the crank-shaft, or if

p = pressure measured on the diagram, in pounds.
A = area of piston, in square inches.
L = length of crank, in inches,

the twisting moment in inch-tons, exerted when the crank makes an angle θ with the line of dead points, is

$$= \frac{pAL}{2240} \frac{\sin(\theta+\phi)}{\cos\phi}.$$

The values of the last term for successive angles of $22\frac{1}{2}°$, when the length of the connecting-rod is twice that of the stroke, which is its usual value in marine engines, are given at p. 650.

The ordinates of the curve of twisting moments, when the obliquity of the connecting-rod is taken into account, may be obtained geometrically as follows:—

The twisting moment is $= Q \times OD$ (Fig. 257)

$$= \frac{P}{\cos\phi} \times OE \cos DOE$$

but by similar triangles, $DOE = DGO = \phi$

$$\therefore \text{ twisting moment} = \frac{P}{\cos\phi} \times OE \cos\phi = P \times OE$$

Consequently, for any angle of the crank, the ordinate of the curve of twisting moments will be proportional to the part OE of the vertical radius intercepted by a line drawn in the direction of the connecting-rod at the instant.

For the corresponding angle of crank, if the connecting-rod were infinite, the twisting moment would be proportional to the vertical dotted line C F.

To obtain the combined twisting moments for the several cylinders of an engine, the curves representing the twisting moments for the respective cylinders are first drawn, the curve for the second cylinder

commencing at the point corresponding to the angle its crank makes with the crank of the first cylinder, and so on. The ordinates of the curve showing the total twisting moment represent the sum of the ordinates of the twisting moments exerted by the several cylinders at the given angles.

An example of this is given in Fig. 258, which shows the twisting moments on the crank-shaft of a two-cylinder compound engine with

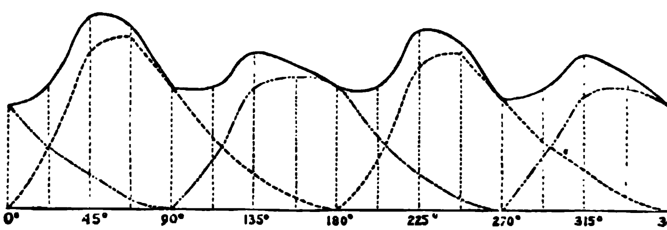


FIG. 258.

cylinders $55\frac{3}{4}$ inches and 98 inches diameter, stroke 2ft. 9in., and length of connecting-rod 5 ft. 6 in. Copies of the indicator diagrams from this engine are given on page 510.

EFFECT OF THE INERTIA OF THE RECIPROCATING PARTS OF THE ENGINES.

In the foregoing examples the pressure of the steam on the piston at any time, as shown by the indicator diagram, has been assumed to represent the pressure on the crank-pin for the corresponding part of the stroke. This, however, will be considerably modified by the inertia of the reciprocating parts of the engines. The angular velocity of the crank-pin is supposed to be uniform, the crank moving through equal angles in equal times. The piston, rods, etc., however, are at rest at the beginning of each stroke, their velocity gradually increases up to mid-stroke, when it reaches a maximum, after which the velocity decreases and becomes zero again at the end of the stroke.

The acceleration during the first half of the stroke can only be produced by the exercise of a pressure, which pressure must evidently be deducted from the steam pressure on the piston in order to obtain the actual pressure on the crank-pin. During the second half of the stroke, when the motion of the piston, etc., is being retarded by the action of the crank-pin, the work accumulated during the acceleration is given out in pressure, which has to be added to the steam pressure on the piston in order to get the total pressure on the crank-pin. The

pressure producing the retardation is practically equal to that producing the acceleration, the only difference being due to the alteration in the velocity of the piston at opposite ends of the stroke resulting from the obliquity of the connecting-rod.

The effect of the inertia of the reciprocating parts, therefore, is to alter the distribution of the pressures on the crank-pin during the stroke; so that whilst, if friction be neglected, the total force that acts on the crank-pin during the stroke is equal to that acting on the piston, yet in the first half of the stroke the pressures on the crank-pin are less, and in the second half greater, than those on the piston. It is therefore clear that the variation of strains on the crank-shaft will be considerably affected from this cause, and it is important that the weights of the reciprocating parts should always be taken into account in constructing the curves of twisting moments.

From the laws of motion we know that if a body of weight W move from rest under the action of a constant accelerating force R, at the end of t seconds

$$\mathrm{R}\,t = \frac{\mathrm{W}}{g}v$$

where v = velocity in feet per second, and $g = 32{\cdot}2$, the accelerating force of gravity.

If s be the space through which the body has moved in the time t, we have

$$v = \frac{2\,s}{t}$$

$$\therefore\ \mathrm{R}\,t = \frac{\mathrm{W}}{g}\cdot\frac{2\,s}{t};\ \text{or}\ \mathrm{R} = \frac{\mathrm{W}}{g}\cdot\frac{2\,s}{t^2}$$

In the case of an engine the acceleration for the first few degrees of the crank is practically uniform, but it soon begins to diminish, and at mid-stroke, when the velocity has reached a maximum, and is for the instant uniform, the acceleration becomes zero. As a matter of fact the acceleration is absolutely greatest at the beginning of the stroke, and diminishes gradually up to half-stroke; though for the first two or three degrees the rate of diminution is so slow that for our present purpose it may be regarded as practically constant for the time under consideration. The difference between velocity and acceleration must be borne in mind. The acceleration is a maximum when the velocity is least, and becomes zero when the velocity reaches its maximum, and is for the instant uniform. This may be clearly seen when we remember that it is only *change of velocity* that requires the exertion of a force, so that, neglecting friction, when a body is moving uniformly no force is required to keep it in motion.

The inertia of a body may be defined as the property it has of

when at rest remaining at rest, or, when in motion continuing to move with uniform velocity unless acted on by some external force.

In the case of the reciprocating parts of the machinery of a steam-engine if we apply the formula

$$R = \frac{W}{g} \cdot \frac{2s}{t^2}$$

we find that for the first degree of revolution, during which the acceleration is practically uniform,

if L = length of crank in feet.

n = number of revolutions *per second.*

$s = L \times \cdot 0001523$; $\cdot 0001523$ being the versed sine of an angle of one degree, if the radius be taken equal to unity.

Also, $t = \dfrac{1}{360\,n}$.

Therefore, the force R necessary to produce the given acceleration in the moving parts

$$= \frac{W}{g} \cdot \frac{2s}{t^2}$$

$$= \frac{W}{g} \cdot \frac{2\,L \times \cdot 0001523}{\dfrac{1}{(360\,n)^2}}$$

$$= 1 \cdot 227 \text{ W.L.} n^2$$

If N = number of revolutions *per minute*,

$$R = \cdot 00034 \text{ W.L.N}^2.$$

If this be divided by the area of the piston in square inches, we shall get the equivalent pressure on the piston to produce the acceleration; which pressure must be deducted from that given by the indicator diagram at the beginning, and added to that given by the diagram at the end of the stroke, in order to obtain the actual pressures exerted on the crank-pin.

If p represents this pressure, and A = area of the piston,

$$p = \frac{R}{A}$$

$$= 1 \cdot 227 \frac{W\,L\,n^2}{A}$$

$$\text{or} = \cdot 00034 \frac{W\,L\,N^2}{A}$$

In order, therefore, to obtain the pressures on the crank-pin at the respective portions of the stroke, a diagram showing the pressures required to accelerate or retard the reciprocating parts must be combined with the indicator diagram. For practical purposes, the variation of velocity of the piston at the opposite ends of the stroke

produced by the obliquity of the connecting rod, may be neglected. The amount of acceleration will then gradually diminish and become zero at mid-stroke, when retardation commences, and then gradually increases until at the end of the stroke it becomes equal to the acceleration at the beginning of the stroke. The diagram showing the

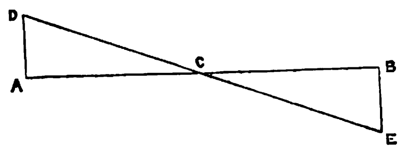


FIG. 259.

accelerating and retarding forces will therefore be similar to Fig. 259.

Let AB represent the length of the indicator diagram.

AD = BE = p = pressure due to acceleration at the beginning, and retardation at the end, of the stroke.

The straight line joining D and E will show the variation in the accelerating forces produced by the inertia of the reciprocating parts throughout the stroke. At C, the centre of the stroke, the acceleration is zero.

The combination of the two diagrams may perhaps be best illustrated by its application to a theoretical diagram. Let A F G H B, Fig. 260, be the indicator diagram; the initial effective pressure on

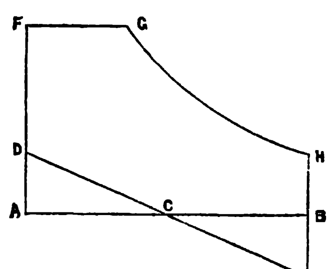


FIG. 260.

the *piston* being A F and the final B H. A D = B E = pressure due to the acceleration of the reciprocating parts, calculated as before explained. Join D E. Then the effective pressure on the *crank-pin* at the beginning of the stroke is D F, and at the end H E. The varying pressures on the crank-pin during the stroke are given by the diagram D F G H E D, and are evidently much more uniform than those given by the indicator diagram A F G H B, which shows the steam pressures on the piston only. At the middle of the stroke

the pressures on the piston and on the crank-pin are equal to each other.

In the case of an ordinary indicator diagram, the form of which is much less regular than that of the theoretical diagram, in order to obtain the pressure on the crank-pin, the acceleration diagram should be drawn as directed, and then the several ordinates of the indicator diagram decreased or increased by the values of the accelerating or retarding forces at the respective parts of the stroke, to form the diagram that gives the pressures on the crank-pin.

Fig. 261 is the diagram of twisting moments on the crank-shaft of the engine referred to at p. 652, when allowance is made for the inertia of the reciprocating parts.

In vertical engines the pressures on the crank-pin are also affected by the dead weight of the moving parts. This is equivalent to the

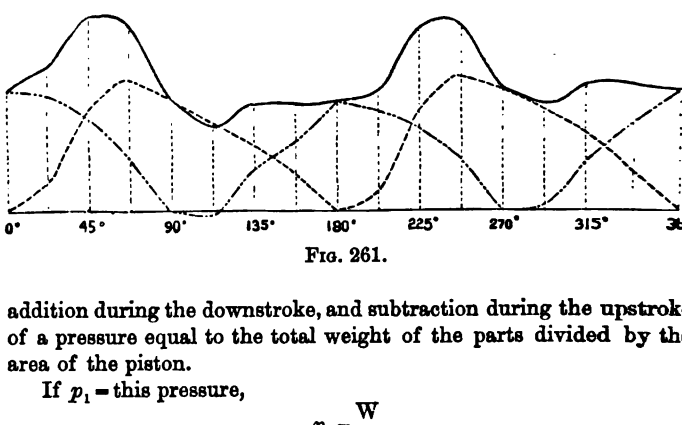


FIG. 261.

addition during the downstroke, and subtraction during the upstroke of a pressure equal to the total weight of the parts divided by the area of the piston.

If p_1 = this pressure,

$$p_1 = \frac{W}{A}$$

and the pressures to be subtracted from or added to the pressures on the indicator diagram will be those given by the acceleration diagram, plus or minus $\frac{W}{A}$, according as the stroke is down or up.

EXTRACTS FROM THE BOARD OF TRADE RULES RELATING TO MACHINERY AND BOILERS.

Iron Boilers.—The Surveyor is to fix the limits of weight to be placed on the safety-valves of passenger steam-ships. In performing this very responsible and onerous duty he must be very careful, as in the event of accident it will be necessary for him to satisfy the Board of Trade that he used due caution. On the one hand he must be careful as regards safety, and on the other hand he must not unduly reduce the pressure on a boiler.

Stays.—In the case of new boilers the Surveyors may allow a stress not exceeding 7,000 lbs. per square inch of net section on solid iron screwed stays supporting flat surfaces, but the stress should not exceed 5,000 lbs. when the stays have been welded or worked in the fire. The areas of diagonal stays are found in the following way:—Find the area of a direct stay needed to support the surface, multiply this area by the length of the diagonal stay, and divide the product by the length of a line drawn at right angles to the surface supported to the end of the diagonal stay; the quotient will be the area of the diagonal stay required. When gusset stays are used their area should be in excess of that found in the above way.

Girders.—When the tops of the combustion boxes or other parts of a boiler are supported by solid rectangular girders the following formula should be used for finding the working pressure to be allowed on the girders, assuming that they are not subjected to a greater temperature than the ordinary heat of steam, and in the case of combustion chambers that the ends are fitted to the edges of the tube-plate, and the back-plate of the combustion box:—

$$\frac{C \times d^2 \times T}{(W - P)\, D \times L} = \text{working pressure.}$$

W = width of combustion box in inches.
P = pitch of supporting bolts in inches.
D = distance between the girders from centre to centre in inches.

U U

L = length of girder in feet.
d = depth of girder in inches.
T = thickness of girder in inches.
C = 500 when the girder is fitted with one supporting bolt.
C = 750 when the girder is fitted with two or three supporting bolts.
C = 850 when the girder is fitted with four supporting bolts.

The working pressure for the supporting bolts and for the plate between them should be determined by the rule for ordinary stays.

Flat Surfaces.—The pressure on plates forming flat surfaces is found by the following formula:—

$$\frac{C \times (T+1)^2}{S-6} = \text{working pressure.}$$

T = thickness of the plate in sixteenths of an inch.
S = surface supported in square inches.
C = constant according to the following circumstances.
C = 100 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts and washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plates they cover.[1]
C = 90 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts only.
C = 60 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, and the stays fitted with nuts and washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plate they cover.
C = 54 when the plates are exposed to the impact of heat or flame, and steam in contact with the plate, and the stays fitted with nuts only.
C = 80 when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the plate and fitted with nuts.

[1] If the diameter of riveted washers be at least two-thirds the pitch of the stays, and the thickness not less than the plates they cover, the constant may be increased to 150.

When doubling-plates are fitted of the same thickness as the plates they cover, and not less in width than two-thirds of the pitch of the stays, the constant may be increased to 160.

When doubling-plates cover the whole of the flat surface the case should be submitted for the consideration of the Board.

C = 60 when the plates are exposed to the impact of heat or flame, with water in contact with the plate, and the stays screwed into the plate, having the ends riveted over to form a substantial head.

C = 36 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, with the stays screwed into the plate, and having the ends riveted over to form a substantial head.

In cases where plates are stiffened by T or L irons, and a greater pressure is required for the plate than is allowed by the use of the above constants, the case should be submitted for the consideration of the Board of Trade.

When the riveted ends of screwed stays are much worn, or when the nuts are burned, the constants should be reduced, but the Surveyor must act according to the circumstances that present themselves at the time of survey, and it is expected that in cases where the riveted ends of screwed stays in the combustion boxes and furnaces are found in this state it will be often necessary to reduce the constant 60 to about 36.

The Surveyors should not in any case allow a greater compressive stress on the tube plates than is obtained by the following formula:—

$$\frac{(D-d)\ T \times 15{,}000}{W \times D.} = \text{working pressure.}$$

D = least horizontal distance between centres of tubes in inches.

d = inside diameter of ordinary tubes in inches.

T = thickness of tube-plate in inches.

W = extreme width of combustion-box in inches from front of tube-plate to back of fire-box, or distance between combustion-box tube-plates when boiler is double-ended and the box common to the furnaces at both ends.

Cylindrical boilers.—The Board of Trade consider that boilers well constructed, well designed, and made of good material should have an advantage in the matter of working pressure over boilers inferior in any of the above respects, as unless this is done the superior boiler is placed at a disadvantage, and good workmanship and material will be discouraged. They have therefore caused the following rules to be prepared:—

When cylindrical boilers are made of the best material with all the rivet holes drilled in place and all the seams fitted with double butt straps each of at least five-eighths the thickness of the plates they cover, and all the seams at least double riveted with rivets having an allowance of not more than 75 per cent. over the single shear, and provided that the boilers have been open to inspection

during the whole period of construction, then 5 may be used as the factor of safety. The tensile strength of the iron is to be taken as equal to 47,000 lbs. per sq. inch with the grain, and 40,000 lbs. across the grain. But when the above conditions are not complied with, the additions in the following scale must be added to the factor 5, according to the circumstances of each case:—

A†	·15	To be added when all the holes are fair and good in the longitudinal seams, but drilled out of place after bending.
B†	·3	To be added when all the holes are fair and good in the longitudinal seams, but drilled out of place before bending.
C	·3	To be added when all the holes are fair and good in the longitudinal seams, but punched *after* bending instead of drilled.
D	·5	To be added when all the holes are fair and good in the longitudinal seams, but punched *before* bending.
E*	·75	To be added when all the holes are not fair and good in the longitudinal seams.
F	·1	To be added if the holes are all fair and good in the circumferential seams, but drilled out of place after bending.
G†	·15	To be added if the holes are fair and good in the circumferential seams, but drilled before bending.
H	·15	To be added if the holes are fair and good in the circumferential seams, but punched after bending.
I†	·2	To be added if the holes are fair and good in the circumferential seams, but punched before bending.
J*	·2	To be added if the holes are not fair and good in the circumferential seams.
K	·2	To be added if double butt straps are not fitted to the longitudinal seams, and the said seams are lap and double riveted.
L	·1	To be added if double butt straps are not fitted to the longitudinal seams, and the said seams are lap and treble riveted.
M	·3	To be added if only single butt straps are fitted to the longitudinal seams, and the said seams are double riveted.
N	·15	To be added if only single butt straps are fitted to the longitudinal seams, and the said seams are treble riveted.
O	1·	To be added when any description of joint in the longitudinal seams is single riveted.
P	·1	To be added if the circumferential seams are fitted with single butt straps and are double riveted.
Q	·2	To be added if the circumferential seams are fitted with single butt straps and are single riveted.

R	·1	To be added if the circumferential seams are fitted with double butt straps and are single riveted.
S††	·1	To be added if the circumferential seams are lap joints and are double riveted.
T	·2	To be added if the circumferential seams are lap joints and are single riveted.
U	·25	To be added when the circumferential seams are lap and the strakes of plates are not entirely under or over.
V‡	·3	To be added when the boiler is of such a length as to fire from both ends, or is of unusual length, such as flue boilers; and the circumferential seams are fitted as described opposite P, R, and S, but, of course, when the circumferential seams are as described opposite Q and T, V ·3 will become V ·4.
W*	·4	To be added if the seams are not properly crossed.
X*	·4	To be added when the iron is in any way doubtful, and the Surveyor is not satisfied that it is of the best quality.
Y†††	1·65	To be added if the boiler is not open to inspection during the whole period of its construction.

Where marked * the allowance may be increased still further if the workmanship or material is very doubtful or very unsatisfactory.

† When the holes are to be rimered or bored out in place the case should be submitted to the Board as to the reduction or omission of A, B, G, and I as heretofore.

‡ When boilers are comparatively short the cases should be submitted to the Board for consideration as to the omission of V as heretofore.

†† When the circumferential seams are lapped and treble-riveted the case should be submitted to the Board.

††† When surveying boilers that have not been open to inspection during construction, the case should be submitted to the Board as to the factors to be used.

The strength of the joints is found by the following method:—

$$\frac{(\text{Pitch} - \text{Diameter of rivet}) \times 100}{\text{Pitch.}} = \begin{cases}\text{Percentage of strength of} \\ \text{plate at joint as compared} \\ \text{with the solid plate.}\end{cases}$$

$$\frac{(\text{Area of rivet} \times \text{No. of rows of rivets}) \times 100}{\text{Pitch} \times \text{thickness of plate.}} = \begin{cases}\text{Percentage of strength} \\ \text{of rivets as compared} \\ \text{with the solid plate.}^{1}\end{cases}$$

Then take iron as equal to 47,000 lbs. per square inch and use

[1] If the rivets are exposed to double shear multiply the percentage as found by 1·75.

the smaller of the two percentages as the strength of the joint, and adopt the factor of safety as found from the preceding scale:

$$\frac{(47{,}000 \times \text{percentage of strength of joint}) \times \text{twice the thickness of the plate in inches.}}{\text{Inside diameter of the boiler in inches} \times \text{factor of safety}}$$

= Pressure to be allowed per square inch on the safety valves.

In the case of zigzag riveting the strength through the plate diagonally between the rivets is equal to that horizontally between the rivets, when diagonal pitch = $\frac{6}{10}$ horizontal pitch + $\frac{4}{10}$ diameter of rivet.

Plates that are drilled in place must be taken apart and the burr taken off, and the holes slightly countersunk from the outsides.

Butt straps.—Butt straps must be cut from plates and not from bars, and must be of as good a quality as the shell plates, and for the longitudinal seams must be cut across the fibre. The rivet holes may be punched or drilled when the plates are punched or drilled OUT of place, but when drilled in place must be taken apart and the burr taken off and slightly countersunk from the outside. When single butt straps are used and the rivet holes in them punched they must be one-eighth thicker than the plates they cover.

The diameter of the rivets must not be less than the thickness of the plates of which the shell is made, but it will be found when the plates are thin, or when lap joints or single butt straps are adopted, that the diameter of the rivets should be in excess of the thickness of the plates.

Dished ends must be stayed as flat surfaces; but when they are theoretically equal to the pressure needed, when considered as portions of spheres, the stays, when solid, may have a stress of 14,000 lbs. per square inch of net section, but the stress should not exceed 10,000 lbs. when the stays have been welded or worked in the fire. Truly hemispherical ends subjected to internal pressure may be allowed double the pressure that is suitable for a cylinder of the same diameter and thickness.

All man-holes and openings must be stiffened with compensating rings of at least the same effective sectional area as the plates cut out, and in no case should the plate rings be less in thickness than the plates to which they are attached. The openings in the shells of cylindrical boilers should have their shorter axes placed longitudinally. It is very desirable that the compensating rings round openings in flat surfaces be made of L or T iron. Cast-iron doors are not to be passed.

The neutral part of boiler shells under steam domes must be efficiently stiffened and stayed, as serious accidents have arisen from the want of such precautions. The boilers must be tested by hydraulic pressure to twice the working pressure in the presence and to the satisfaction of the Board's Surveyors.

Circular furnaces.—Circular furnaces with the longitudinal joints welded or made with a butt strap :—

$$\frac{90{,}000 \times \text{the square of the thickness of the plate in inches}}{(\text{Length in feet} + 1) \times \text{diameter in inches}}$$

= working pressure per square inch, provided it does not exceed that found by the following formula :—

$$\frac{8{,}000 \times \text{thickness in inches}}{\text{Diameter in inches}} = \text{working pressure per square inch.}$$

The second formula limits the crushing stress on the material to 4,000 lbs. per square inch. The length is to be measured between the rings if the furnace is made with rings.

If the longitudinal joints instead of being butted are lap-jointed in the ordinary way, then 70,000 is to be used instead of 90,000, excepting only where the lap is bevelled and so made as to give the flues the form of a *true* circle, when 80,000 may be used. When the material or the workmanship is not of the best quality, the constants given above must be reduced, that is to say—the 90,000 will become 80,000; the 80,000 will become 70,000; the 70,000 will become 60,000. When the material and the workmanship are not of the best quality, such constants will require to be further reduced, according to circumstances and the judgment of the Surveyor, as in the case of old boilers. One of the conditions of best workmanship is, that the joints are either double riveted with single butt straps, or single riveted with double butt straps, and the holes drilled after the bending is done and when in place, and the plates afterwards taken apart, the burr on the holes taken off, and the holes slightly countersunk from the outside.[1]

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[1] The following examples will serve to show the application of the constants for the different cases that may arise :—

Furnaces with butt joints and drilled rivet holes:
- 90,000 where the longitudinal seams are welded.
- 90,000 where the longitudinal seams are double riveted and fitted with single butt straps.
- 80,000 where the longitudinal seams are single riveted and fitted with single butt straps.
- 90,000 where the longitudinal seams are single riveted and fitted with double butt straps.

The working pressure for corrugated furnaces, practically circular and machine made, provided the plane parts at the ends do not exceed 6 inches in length, and the plates are not less than $\frac{5}{16}$-inch thick, should not be greater than found by the following formula:—

$$\frac{9{,}000 \times \text{thickness in inches}}{\text{Mean diameter in inches}} = \text{working pressure per square inch.}$$

When the furnaces are riveted in two or more lengths the case should be submitted for consideration, as it may be necessary to make a deduction.

Steel Boilers.—The following should guide the Board's Surveyors when the general quality of the steel has been found suitable for marine boilers.

Tests.—The steelmakers or boilermakers should test one or more strips cut from *each* plate for tensile strength and elongation, and stamp both results on a part of the plate where they can be easily seen when the boiler is constructed. The Surveyor is not obliged to witness the foregoing tests, although it is very desirable that he should when his other duties will allow him to do so, but he should, however, select at least one in four of these plates, either at the steel works or the boilermakers' works, and witness the testing of at least one strip cut from each selected plate.

Furnaces with butt joints and punched rivet holes	85,000	where the longitudinal seams are double riveted and fitted with single butt straps.
	75,000	where the longitudinal seams are single riveted and fitted with single butt straps.
	85,000	where the longitudinal seams are single riveted and fitted with double butt straps.
Furnaces with lapped joints and drilled rivet holes	80,000	where the longitudinal seams are double riveted and bevelled.
	75,000	where the longitudinal seams are double riveted and not bevelled.
	70,000	where the longitudinal seams are single riveted and bevelled.
	65,000	where the longitudinal seams are single riveted and not bevelled.
Furnaces with lapped joints and punched rivet holes	75,000	where the longitudinal seams are double riveted and bevelled.
	70,000	where the longitudinal seams are double riveted and not bevelled.
	65,000	where the longitudinal seams are single riveted and bevelled.
	60,000	where the longitudinal seams are single riveted and not bevelled.

In the case of upright fireboxes of donkey or similar boilers, 10 per cent. should be deducted from the constants given above, applicable to the respective classes of work.

A few rivets of each size should be selected, and should be turned and tested for tensile stress. The tensile stress should be from 28 tons to 32 tons per square inch, with a contraction of area of about 60 per cent.

If for the plates from which the Surveyor selects the above proportions a greater stress is wished than is allowed for iron, tests for tensile stress and elongation should be made, and those for which no reduction of thickness is asked may be tested for resistance to bending, if preferred. In the latter case the tensile stress and elongation stamped on each plate should be reported by the Surveyor to the Board of Trade, along with the results of the bending tests.

The breadth of the test strips for tensile stress should be about 2 inches, and the elongation, taken in a length of 10 inches, should be about 25 per cent., and not less than 20 per cent. The test strips must be carefully prepared and measured, and they should be cut from the plate by a planing or shaping machine.

The bending tests for the plates *not* exposed to flame should be made with strips in their normal condition, but strips cut from furnaces, combustion boxes, &c. should be heated to a cherry red, then plunged into water of about 80° and kept there until of the same temperature as the water, and then bent. The bending strips should not be less than 2 inches broad and 10 inches long, and they should be bent until they break, or until the sides are parallel at a distance from each other of not less than (3) three times the thickness of plate.

The tensile stress of the plates *not* exposed to flame should be about 28 tons, and should not exceed 32 tons, per square inch of section, and 28 tons should be the stress used in the calculations for cylindrical shells if the plates comply with all the conditions as stated herein, but when the minimum tensile stress of shell plates exceeds 28 tons and allowance is wished for the excess, the case should be specially submitted for the consideration of the Board. The tensile stress of furnace, flanging, and combustion box plates should range from 26 tons to 30 tons per square inch.

All plates that are punched, flanged, or locally heated must be carefully annealed after being so treated.

The rivet holes in the furnaces and longitudinal seams of cylindrical shells should be *drilled*, but if it is wished to punch them and afterwards bore or anneal the plates in a proper furnace the particulars of the punching and boring or annealing should be submitted to the Board of Trade for consideration before being done, but all punched holes should be made after bending.

In all cases where assent has been given for plates to be punched

after bending and then annealed, the makers should stamp the plates with the words 'punched after bending and then annealed,' and in all cases where assent has been given for punching and afterwards boring plates the words 'punched and then bored' should be stamped on the plates.

Stays.—Bars for stays should be tested. Solid steel screwed stays which have *not* been welded or otherwise worked after heating may be allowed a working stress of 9,000 lbs. per square inch of net section, provided the tensile stress is from 28 to 32 tons per square inch, and the elongation in 10 inches about 25 per cent. and not less than 20 per cent. Steel stays which have been welded or worked in the fire have been found to be unreliable, therefore they should *not* be passed.

Flat Surfaces.—If the flanging plates and those exposed to flame comply with the foregoing conditions, the constants in the Board's rules for iron boilers may be increased as follows:—

The constants for flat surfaces when they are supported by stays screwed into the plate and riveted, 10 per cent. The constants for flat surfaces when they are supported by stays screwed into the plate and nutted, or when the stays are nutted in the steam space, 25 per cent. This is also applicable to the constants for flat surfaces stiffened by riveted washers or doubling strips, and supported by nutted stays. The constants for combustion-box girders 10 per cent.

Furnaces.—The furnace constants about 10 per cent. when plane. When new, corrugated and machine-made and practically true circles, the working pressure is found by the following formula, provided that the plane parts at the ends do not exceed 6 inches in length, and the plates are not less than $\frac{5}{16}$-inch thick,

$$\frac{12,500 \times T}{D} = \text{working pressure,}$$

T = thickness in inches.
D = mean diameter in inches.

(If the furnace is riveted in two or more lengths, the case should be submitted for consideration.)

A greater compressive stress should not be allowed on tube plates than that found by the following formula:—

$$\frac{(D - d)\ T \times 20,000}{W \times D} = \text{working pressure.}$$

D = least horizontal distance between centres of tubes in inches.
d = inside diameter of tubes in inches.
T = thickness of tube plate in inches.

W = extreme width of combustion-box in inches from front of tube plate to back of fire-box, or distance between combustion-box tube plates when boiler is double-ended and the box common to the furnaces at both ends.

The rivet section, if of iron, in the longitudinal seams of cylindrical shells where lapped and at least double riveted should not be less than $\frac{13}{8}$ times the net plate section, but if steel rivets are used their section should be at least $\frac{28}{23}$ of the net section of the plate if the tensile stress of the rivets is not less than 28 tons and not more than 32 tons per square inch. Therefore, in calculating the working pressure, the percentage strength of the rivets may be found in the usual way by the Board's rules; but in the case of iron rivets the percentages found should be divided by $\frac{13}{8}$, and in the case of steel rivets by $\frac{28}{23}$, the results being the percentages required. If the percentage strength of the rivets by calculation is less than the calculated percentage strength of the plate, calculate the working pressure by both percentages. When using the percentage strength of the plate use the nominal factor of safety suitable for the method of construction as by the Board's rules for iron boilers, but when using the percentage strength of the rivets use 5 as the factor of safety. The less of the two pressures so found is the working pressure to be allowed for the cylindrical portion of the shell.

Local heating of the plates should be avoided, as many plates have failed from having been so treated. Steel plates which have been welded should not be passed if subject to a tensile stress, and those welded and subject to a compressive stress should be efficiently annealed. In other respects the boilers should comply with the rules for iron boilers.

Safety valves.—The provisions relating to safety valves are in substance as follows:—Every steam-ship of which a survey is required shall be provided with a safety valve upon each boiler, so constructed as to be out of the control of the engineer when the steam is up, and if such valve is in addition to the ordinary valve, it shall be so constructed as to have an area not less, and a pressure not greater, than the area of and pressure on that valve.

Cases have come under the notice of the Board of Trade in which steam-ships have been surveyed, and passed by the Surveyors with pipes between the boilers and the safety-valve chests. Such arrangement is not in accordance with the Act, which distinctly provides that the safety valves shall be upon the boilers.

The area per square foot of fire-grate surface of Government safety valves, or (when there is more than one Government safety

valve on the boiler) the combined area of the Government safety valves locked up should be not less than that given in the following tables opposite the boiler pressure intended, provided the valves are not less than three inches in diameter. When, however, the valves are of the common description, and are made in accordance with the tables, it will be necessary to fit them with springs having great elasticity, or to provide other means to keep the accumulation within moderate limits; and as boilers with forced draught require valves considerably larger than those found by the tables, the design of the valves proposed for such boilers should be submitted to the Board for consideration.

SAFETY VALVE AREAS.

Boiler Pressure	Area of Valve per square foot of Fire-grate	Boiler Pressure	Area of Valve per square foot of Fire-grate
lbs.	sq. in.	lbs.	sq. in.
60	·500	98	·331
62	·487	100	·326
64	·474	105	·312
66	·462	110	·300
68	·451	115	·288
70	·441	120	·277
72	·431	125	·267
74	·421	130	·258
76	·412	135	·250
78	·403	140	·241
80	·394	145	·234
82	·386	150	·227
84	·378	155	·220
86	·371	160	·214
88	·364	170	·202
90	·357	180	·192
92	·350	190	·182
94	·344	200	·174
96	·337		

The safety valves should be fitted with lifting gear, so arranged that the two or more valves on any one boiler can at all times be eased together, without interfering with the valves on any other boiler. The lifting gear should in all cases be arranged so that it can be worked by hand either from the engine-room or stokehole.

Care should be taken that the safety valves have a lift equal to at least one-fourth their diameter; that the openings for the passage of steam to and from the valves, including the waste steam pipe, should each have an area not less than the area of the valves,

and that each valve-box has a drain-pipe fitted at its lower part. In the case of lever valves, if the lever is not bushed with brass, the pins must be of brass; iron and iron working together must not be passed. Too much care cannot be devoted to seeing that there is proper lift, and free means of escape of waste steam, as it is obvious that unless the lift and means for escape of waste steam are ample, the effect is the same as reducing the area of the valve or putting on an extra load. The valve seats should be secured by studs and nuts.

Size of shafting.—Main, and tunnel, and propeller shafts must not be passed if found to be less in diameter than that found by the following rules, without previously submitting the whole case to the Board of Trade for their consideration. It will be found that first-class makers generally put in larger shafts than those found by the following formulæ.

For compound engine with two cylinders:

$$\text{Diameter of shaft in inches} = \sqrt[3]{\frac{(d^2 \times P) + (D^2 \times 15)}{f} C},$$

where d = diameter of high-pressure cylinder in inches.
D = diameter of low-pressure cylinder in inches.
P = boiler pressure.
C = length of crank in inches.
f = constant from following table.

For ordinary condensing engines, with two cylinders, when the pressure is not low:

$$\text{Diameter of shaft in inches} = \sqrt[3]{\frac{D^2 \times P \times 2}{f} C},$$

where D = diameter of cylinder.
P = boiler pressure.
C = length of crank in inches.
f = constant from following table.

Coustant	Angle between Cranks	For Crank and Propeller Shafts	For Tunnel Shaft
f	90°	2468	2880
,,	100°	2279	2659
,,	110°	2131	2487
,,	120°	2016	2352
,,	130°	1926	2248
,,	140°	1858	2168
,,	150°	1806	2108
,,	160°	1772	2068
,,	170°	1752	2045
,,	180°	1746	2037

LLOYD'S RULES FOR BOILERS.

Steel Boilers.—The use of steel will be sanctioned in the construction of boilers intended for vessels classed or proposed for classification in the Society's Register Book, provided the boilers be constructed in accordance with the requirements of the Rules, and the following conditions be fulfilled.

The material is to have an ultimate tensile strength of not less than 26 and not more than 30 tons per square inch of section,[1] with an ultimate elongation of not less than 20 per cent. in a length of eight inches. It is to be capable of being bent to a curve of which the inner radius is not greater than one and a half times the thickness of the plates or bars, after having been heated uniformly to a low cherry-red and quenched in water of 82 degrees Fahrenheit.

Steel rivets are to be considered as part of the material, and in addition to being subjected to a shearing test, they must be capable of withstanding the same tests as the plates are required to undergo.

Samples for testing are to be selected from each batch of plates submitted for approval, care being taken in the selection that, as far as possible, each cast or furnace charge from which the material has been produced is represented.[2] In addition to these tests, the temper test is to be applied to samples taken from *every* plate intended to be used in the furnaces and combustion chambers of the boilers.

The Society's Surveyor will attend at the steel works when necessary, and select the samples for testing before the plates are

[1] Steel of a less tensile strength than 26 tons per square inch, if satisfactory in other respects, may be allowed in any case where the scantlings are equal to those prescribed in the Rules for Iron Boilers. In such cases the Surveyors should represent the facts for the Committee's consideration.

[2] When a great number of charges are represented in the number of plates submitted, a proportion of one tensile and one temper test to every ten plates will be deemed sufficient, provided they all prove to be satisfactory.

sheared to size, and these samples when marked by him for testing should, as far as practicable, be followed by the Surveyor through the different stages of preparation until the tests are completed.

The Society's Surveyor will require to have every facility placed in his way for tracing all plates to their respective charges, and to be furnished with two copies of the advice notes of the material, one of which, when he shall have been satisfied with the results of the tests applied to the material, is to be signed and forwarded to the boiler manufacturer, and the other is to be retained by himself.

The samples are taken for testing in order that the general quality of the material may be ascertained, and if any sample should fail to fulfil the conditions laid down, the plate from which the sample is taken must be rejected; and further tests should be made before any material, made from the same cast or charge as the failing sample, can be approved.

All the holes in steel boilers should be drilled; but if they be punched, the plates are to be afterwards annealed.

All plates that are dished or flanged, or in any way heated in the fire for working, except those that are subjected to a compressive stress only, are to be annealed after the operations are completed. No steel stays are to be welded.

Unless otherwise specified, the Rules for the construction of iron boilers will apply equally to boilers made of steel.

Cylindrical Shells.—The strength of circular shells to be calculated from the strength of the longitudinal joints by the following formula:—[1]

$$\frac{C \times T \times B}{D} = \text{working pressure}$$

where C = coefficient as per following table.
T = thickness of plate in inches.
D = mean diameter of shell in inches.
B = percentage of strength of joint found as follows—the least percentage to be taken.

For plate at joint, $B = \frac{p - d}{p} \times 100$

For rivets at joint, $B = \frac{n \times a}{p \times T} \times 100$ with iron rivets in iron plates with punched holes.

$B = \frac{n \times a}{p \times T} \times 90$ with iron rivets in iron plates with drilled holes.

[1] In any case where the strength of the longitudinal joint is satisfactorily shown by experiment to be greater than that given by this formula, the actual strength may be taken in the calculation.

$$B = \frac{n \times a}{p \times T} \times 85 \text{ with steel rivets in steel plates.}$$

$$B = \frac{n \times a}{p \times T} \times 70 \text{ with iron rivets in steel plates.}$$

(In case of rivets being in double shear, $1{\cdot}75a$ is to be used instead of a.)

where p = pitch of rivets.
d = diameter of rivets.
a = sectional area of rivets.
n = number of rows of rivets.

Table of Coefficients.

Iron Boilers.

Description of Longitudinal Joint	For Plates ½-inch thick and under	For Plates ¾-in. thick and above ½-inch	For Plates above ¾-in. thick.
Lap joint, punched holes	155	165	170
Lap joint, drilled holes .	170	180	190
Double butt strap joint, punched holes.	170	180	190
Double butt strap joint, drilled holes.	180	190	200

Steel Boilers.

Description of Longitudinal Joint	For Plates ⅜-in. thick and under	For Plates 7/16-in. thick and above ⅜	For Plates ¾-in. thick & above 7/16	For Plates above ¾-in. thick
Lap joints	200	215	230	240
Double butt strap joints .	215	230	250	260

Note.—The Inside Butt Strap to be at least three-quarters the thickness of the plate.

Note.—For the shell plates of superheaters or steam chests enclosed in the uptakes or exposed to the direct action of the flame, the coefficients should be two-thirds of those given in the above tables.

Proper deductions are to be made for openings in shell.

All manholes in circular shells to be stiffened with compensating rings.

The shell plates under domes in boilers so fitted, to be stayed from the top of the dome or otherwise stiffened.

Stays.—The strength of stays supporting flat surfaces is to be calculated from the weakest part of the stay or fastening, and the strain upon them is not to exceed the following limits, namely :—

Iron Stays.—For screw stays, and for other stays not exceeding

$1\frac{1}{2}$ inches effective diameter, and for all stays which are welded, 6,000 lbs. per square inch; for unwelded stays above $1\frac{1}{2}$ inches effective diameter, 7,500 lbs. per square inch.

Steel Stays.—For screw stays, and for other stays not exceeding $1\frac{1}{2}$ inches effective diameter, 8,000 lbs. per square inch; for stays above $1\frac{1}{2}$ inches effective diameter, 9,000 lbs. per square inch. No steel stays are to be welded.

Flat Plates.—The strength of flat plates supported by stays to be taken from the following formula :—

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch}$$

where T = thickness of plate in sixteenths of an inch.
P = greatest pitch in inches.
C = 90 for plates $\frac{7}{16}$ thick and below fitted with screw stays with riveted heads.
C = 100 for plates above $\frac{7}{16}$ fitted with screw stays with riveted heads.
C = 110 for plates $\frac{7}{16}$ thick and under fitted with screw stays and nuts.
C = 120 for plates above $\frac{7}{16}$ fitted with screw stays and nuts.
C = 140 for plates fitted with stays with double nuts.
C = 160 for plates fitted with stays with double nuts, and washers at least half thickness of plates and a diameter of $\frac{2}{5}$ of the pitch, riveted to the plates.

NOTE.—In the case of front plates of boilers in the steam space, these numbers should be reduced 20 per cent., unless the plates are guarded from the direct action of the heat.

Girders.—The strength of girders supporting the tops of combustion-chambers and other flat surfaces, to be taken from the following formula :—

$$\frac{C \times d^2 \times T}{(L-P) \times D \times L} = \text{working pressure in lbs. per square inch}$$

where L = length of girder.
P = pitch of stays.
D = distance apart of girders.
d = depth of girder at centre.
T = thickness of girder at centre. All these dimensions to be taken in inches.

$$C = \begin{cases} 6{,}000, \text{ if there is one stay to each girder.} \\ 9{,}000, \text{ if there are two or three stays to each girder.} \\ 10{,}200, \text{ if there are four stays to each girder.} \end{cases}$$

Circular Furnaces.—The strength of furnaces to resist collapsing to be calculated from the following formula:—

$$\frac{89{,}600 \times T^2}{L \times D} = \text{working pressure in lbs. per square inch}$$

where 89,600 = constant.
T = thickness of plates in inches.
D = outside diameter of furnace in inches.
L = length of furnaces in feet. If rings are fitted the length between rings to be taken.

The pressure in no case to exceed $\frac{8{,}000 \times T}{D}$

Fox's corrugated furnaces.—The strength of corrugated furnaces (corrugations $1\frac{1}{2}$ inches deep) to be calculated from the following formula:—

$$\frac{1{,}000 \times (T - 2)}{D} = \text{working pressure in lbs. per square inch;}$$

where T = thickness of plates in sixteenths of an inch,
and D = greatest diameter of furnace in inches.

Safety valves.—Two safety valves to be fitted to each boiler and loaded to the working pressure in the presence of the Surveyor. If common valves are used, their combined areas to be at least half a square inch to each square foot of grate surface. If improved valves are used, they are to be tested under steam in the presence of the Surveyor: the accumulation in no case to exceed 10 per cent. of the working pressure. Each valve to be arranged so that no extra load can be added when steam is up, and to be fitted with easing gear which must lift the valve itself. All safety-valve spindles to extend through the covers and to be fitted with sockets and cross handles, allowing them to be lifted and turned round in their seats, and their efficiency tested at any time.

EXAMINATION QUESTIONS, SELECTED PRINCIPALLY FROM ENGINEERING PAPERS SET FOR ENGINEER STUDENTS, AND OFFICERS STUDYING AT THE ROYAL NAVAL COLLEGE.

1. What are the several elements that constitute the *total efficiency* of the marine steam-engine?

2. Describe briefly the effects produced by the communication of heat to water.

3. What is meant by the 'total heat of evaporation'? Calculate the total heat of evaporation of 20 pounds of steam at a temperature of 300° Fahr.

4. What is meant by the 'total heat of combustion'? Calculate the total heat of combustion, and the theoretical evaporative power of a pound of coal containing 85 per cent. of carbon, 5 per cent. of hydrogen, 5 per cent. of oxygen, and 5 per cent. of ash, etc. What quantity of air would be required for the combustion of a ton of this coal, if natural draught only were used?

5. What causes tend to make the available evaporative power of coal less than its theoretical evaporative power? Give a formula for the efficiency of a boiler; and calculate the efficiency, if the ratio of heating to grate surface is 25 to 1, and 28 pounds of coal are burnt per square foot of grate per hour.

6. State briefly the processes of combustion in an ordinary furnace. What arrangements should be made to render the combustion complete? What is meant by the efficiency of the boiler, on what does it depend, and what is its amount in ordinary cases?

7. Name the various cocks, valves, and other mountings usually fitted to a marine boiler, and describe their uses.

8. How have boilers been constructed to carry steam of very high pressures, say from 200 to 300 pounds per square inch? Give sketches and description of any boiler of the kind with which you are acquainted.

9. Sketch a cylindrical high-pressure boiler showing the mode of construction and staying, and describe the several processes

employed in its construction from the commencement until its completion in the shop.

10. Give sketches showing the different methods of construction of cylindrical furnaces, and state the advantages resulting from the adoption of corrugated furnaces for high-pressure boilers.

11. Sketch and describe the usual kind of spring-loaded safety valve fitted to marine boilers. Why are spring-loaded safety valves preferred to those loaded with weights? How many valves are usually fitted to each boiler, and how should the total area of the safety valves be determined?

12. For what purpose are stop valves placed on boilers? What advantage is gained by making them self-closing? Make a sketch of a self-closing steam stop valve, showing how the internal and external steam pipes are attached to it, and how expansion is allowed for in the external pipe. What provision is made to prevent the pipe being drawn out of the gland?

13. What are the principal causes of deterioration in marine boilers? Name the parts that are liable to deteriorate most rapidly; and state the measures that should be taken when boilers are in use and when not in use to prevent deterioration?

14. How can the rate of combustion of coal in ordinary marine boilers be regulated, and how can it be forced? Describe the arrangements adopted for closing the stokeholds and placing them under forced draught, and state the advantages of this system of working for war-ships.

15. What do you understand by *priming* in boilers? State the chief causes of priming: and the indications given at the engines when priming occurs in the boilers. What precautions are required when the priming is excessive?

16. Describe the usual apparatus fitted in the main steam-pipe and on the cylinders for the purpose of obviating the risk of injury to the machinery from priming; and make a sketch of a separator, showing the internal and external fittings, stating the material of which the several parts are made.

17. What are the uses of Kingston's valves, and where are they fitted? What are the usual tests to which the valve-spindles are subjected? Make a sketch of a Kingston valve, and show how it is attached to the ship in (1) an iron ship with a double bottom, and (2) a composite or a wood ship.

18. Sketch and explain clearly how the furnace of a cylindrical boiler is constructed, how it is attached to the combustion chamber and front plates, and also how expansion is allowed for. How is the operation of flanging such a furnace performed in the shop? Why is great care required in doing this? What special

qualities should the material employed in the construction of furnaces possess, and for what reasons?

19. In order to ascertain the capability of boilers for further service, and their probable durability, what tests are usually employed, and what precautions are necessary in conducting these tests?

20. State the seams in a cylindrical high-pressure boiler, that are usually single, double, and treble riveted respectively. Give reasons why this method of construction is adopted. Make full-sized sketches of a portion of a longitudinal joint of the shell plates and a portion of the joint in the plates in the combustion-chamber of a boiler 10 feet in diameter designed for a working steam pressure of 130 pounds per square inch. Give as many figured dimensions as you can.

21. What are the principal points to be kept in view in arranging for the ventilation of the stokehold: and how is this object usually attained? What special fittings are required for working with forced draught?

22. Give detailed sketches of a furnace door and dead-plate, showing the arrangements for permitting the air to enter above and below the fires for combustion of the gases and prevention of smoke.

23. Enumerate the different classes of boilers employed on board war-ships, and describe their chief distinctive features. Give sketches of each of these types, and state the average and greatest pressures at which they are worked. How are they secured to the ship?

24. What amount of grate and tube surface should be allowed for a cylindrical boiler of 350 I.H.P., with natural draught worked at a steam pressure of 100 pounds per square inch above the atmosphere? Suppose this boiler to be fitted with a pair of spring-loaded safety valves, what should be the diameter of the valves and of what dimensions would you make the springs? What would be the lift of the valves required to liberate all the steam the boilers would generate, when not forced by the application of artificial draught?

25. What is meant by the 'efficiency of the steam,' and on what does it depend? If the initial pressure of the steam at the cylinders, as shown by the gauge, be 60 pounds per square inch, the vacuum in the condenser 26 inches, and the mean pressure, shown by the indicator diagram, be 25 pounds per square inch, find the rate of expansion, final pressure, and the ratio in which the efficiency of the steam has been increased.

26. Suppose a volume of saturated steam to be contained in a non-conducting cylinder fitted with a non-conducting piston.

Explain the results that would follow in the two undermentioned cases :—(*a*) If the piston be moved by an external force so that the steam expands without performing any mechanical work. (*b*) If the piston be pressed forward by the steam against an external pressure so that the steam does mechanical work during its expansion.

27. Give a short description of the manner in which heat passes from the steam to the cylinder and back again to the steam during each stroke in an unjacketed cylinder. How are the losses, due to the initial liquefaction at high rates of expansion, partially prevented by the adoption of the steam jacket? Explain fully.

28. What is meant by superheated steam, and why is economy obtained by superheating? To what extent may superheating be safely carried, and why are superheaters now seldom fitted to high-pressure boilers?

29. Describe the action of the steam in a simple expansion engine, and also in a compound engine, and give sketches of indicator diagrams in illustration of your answers.

30. Whether is it more necessary for the sake of economy of fuel to steam-jacket the cylinders of an ordinary engine or those of a compound engine, the initial pressure and the total rate of expansion of steam being the same in both cases? Give reasons for your answer.

31. Give a sketch of an indicator diagram from a simple expansion engine, explaining by means of it the action of the steam in the cylinder, and give sketches also of the diagrams you would expect to obtain if the engines were out of order from various causes.

32. Give sketches of indicator diagrams from a compound engine with one high-pressure and one low-pressure cylinder, and the cranks at right angles, and show how you would calculate the horse-power of the engine from the diagrams.

33. The indicator diagram of a simple expansion engine shows that the initial pressure of steam was 30 pounds above and the pressure at release 5 pounds below the atmospheric line, back pressure 3 pounds. What would be the mean pressure, and how much would the efficiency of the steam be increased by expansion?

34. Trace the course of the steam from the boilers to the condenser in a compound and also in a triple expansion engine. Enumerate the pipes and valves through which it has to pass in each case.

35. Define the following terms: *latent heat*, *specific heat*, *thermal unit*, *foot-pound*. How many thermal units are required to change 1 lb. of water at 60° Fahr. into steam of atmospheric pressure?

36. When boilers supply steam for engines with surface condensers, what causes prevent the return to the boilers of all the water evaporated, and how is any deficiency of feed-water made up?

37. What was the principle James Watt adopted to increase the efficiency of the steam-engine? State the principal improvements he introduced.

38. When the working pressure of the steam is reduced, owing to weakness of or defects in the boilers, why is it considered necessary to reduce the pitch of the screw at the same time?

39. Why is *blowing out* necessary with jet condensers? Obtain the percentage of loss of heat by blowing out when the water in the boiler is kept at three times the density of sea-water, assuming the temperature of the steam to be 275° Fahr.

40. The area of the piston of an engine is 4,000 square inches, stroke 4 ft., number of revolutions per minute, 90. The initial absolute pressure in the cylinders is 45 lbs. cut-off at one-third stroke. Supposing a jet condenser is fitted, calculate the quantity of condensing water required per hour to give a temperature of 100° in the hot-well. Take the temperature of the sea-water to be 60° Fahr. If a surface condenser be fitted, what quantity of circulating water would be required per hour, supposing the circulating water to be raised 20° in temperature in passing through the condenser?

41. How are the difficulties attending the use of a high rate of expansion with high-pressure steam overcome by the adoption of the compound engine? State fully your views respecting the relative merits of simple, double, and triple expansion engines for ships of war.

42. In what manner is a gain realised by the substitution of surface for jet condensation? Show what is the gain in any particular case, and state why it is necessary to use surface condensation with high-pressure steam.

43. State and explain the several causes that may produce an imperfect vacuum in surface and jet condensers respectively.

44. What is the office of the expansion valve? State the advantages resulting from its use, and give sketches of any expansion valve which you have seen fitted to engines of modern construction.

45. Sketch and describe the ordinary gridiron expansion valve. What objections are there to its use when high rates of expansion have to be carried out? How is this disadvantage in working shown on the indicator diagram, and what precautions should be taken to reduce this to a minimum?

46. Sketch and describe any expansion gear with which you are acquainted.

47. Sketch a locomotive or single-ported slide-valve in the position it should occupy on the cylinder face when the piston is about to commence its stroke, trace the movements of the valve as the piston

moves through a complete stroke, and explain, in order, the various operations performed.

48. Sketch and describe a double-ported slide-valve, and state its advantages. Explain how slide-valves are set practically.

49. Sketch any form of cylindrical or piston slide-valve with which you are acquainted and state the conditions in which this form of valve is desirable.

50. The motion of the slide-valve can be represented by a pair of circles. Draw a valve diagram, and mark upon it the positions of the crank at admission, cut-off, release, and compression, and indicate upon the diagram the outside lap, lead, and half-travel of the slide-valve.

51. Make a sketch of the slide-valve casing, and slide-valve cover of a marine engine, showing the means provided for relieving the pressure from the back of the slide-valve.

52. Explain clearly the meaning of the terms *lap*, *lead*, *angular advance*, *compression*, and *clearance*. For what purpose are the two former applied to slide-valves? How is an ordinary indicator diagram affected by the two latter? How would you proceed to fix the positions on the shaft of the two eccentrics for ahead and astern motion, with respect to the crank?

53. Give a sketch of the link motion with two eccentrics employed to work a slide-valve. Explain its action when going ahead or astern, and when working expansively.

54. Sketch and explain any method with which you are acquainted for reversing an engine fitted with a single eccentric.

55. Sketch and describe fully any plan of slide-valve gear, other than the link motion, with which you are acquainted—state clearly the principle of action of the gear.

56. Explain fully the action of the link motion when used to produce a variable cut-off. What is meant by the *virtual eccentric arm*? Show how this may be found geometrically.

57. Make a sketch of a double-ported slide-valve, with the relief ring on its back. Show also the ports in the cylinder, and the valve in the proper position when the piston is at the end of its stroke.

58. Describe the fittings applied to the cylinders of a set of compound engines for starting, stopping, and reversing. How would you perform these operations?

59. Make a sketch of the section through a cylinder, showing the jacket, liner, and cylinder ports and passages. Describe the usual plan of attaching the liner to the cylinder and of keeping the outer end steam-tight.

60. What fittings are generally applied to the cylinders and pistons of large horizontal engines to prevent them bearing exces-

sively? Give sketches and descriptions of such fittings, and explain how the pistons are lined up to their central positions when they are found to have worn down.

61. Enumerate and describe all the fittings generally attached to the cylinders of a marine steam-engine, and give the reasons why they are necessary.

62. Make a sketch of a steel piston for an engine, showing its metallic packing, the method of fitting the junk ring, and also of attaching the piston to the rod; also show how the joint of the metallic ring is fitted, to prevent the passage of steam.

63. What is understood technically by the term *vacuum*, and how is it produced and measured? How is the recorded vacuum affected by the height of the weather barometer? Explain the construction and operation of an ordinary jet-injection condenser: and describe the action of the air-pump. What is meant by the foot and delivery valves, and of what material are these valves usually made?

64. Describe the construction of a surface condenser, and give sketches. Enumerate the various fittings and their purposes. Under what conditions is an outboard discharge pipe necessary to be fitted to the hot-well of a surface condenser in addition to the circulating pump discharge from the condenser?

65. Describe the action of a surface condenser. Show by rough sketches the relative positions of the air-pump and valves, and the methods of securing the tubes in the tube-plates. Explain the arrangements that would be necessary in a surface condenser and in the discharge pipe of the air-pump, to enable the former to be used as a common condenser, and to prevent the loss of fresh water through the discharge.

66. For the supply of circulating water, which kind of pump is the better—the centrifugal, or the reciprocating worked off the engines? Give reasons. Make a vertical section of a vertical air-pump suitable for inverted engines, showing the passages and valves.

67. If the tubes of a surface condenser are leaking badly what evil effects might happen during a temporary stoppage of the engines, or on starting afterwards? What means are at your disposal to prevent damage? How would you detect the leaky tubes, and how most speedily repair the defect?

68. What amount of cooling surface should be allowed in the surface condenser for a pair of marine engines indicating 1,000 I.H.P.? If the tubes are $\frac{3}{4}$ of an inch in diameter and 5 feet long between the tube-plates, how many tubes would be required? Of what material should the tubes be made?

69. If 20 pounds of steam be used per I.H.P. per hour, how

many tons of condensing water per hour would be required for the engines in the previous question, supposing the difference in temperature of the water in the inlet and outlet orifices to be 20° Fahr.? The temperature of the hot-well may be taken as 100° Fahr.

70. What are meant by the terms *pitch*, *angle*, *leading edge*, and *slip* as applied to screw propellers? What is the difference between the *real* and *apparent slip*? A ship attains a speed of 16 knots per hour, the mean pitch of the screw is 18 feet and the number of revolutions per minute, 88. What is the slip per cent. and is it positive or negative?

71. Show by sketches the mode of attaching a propeller blade to the boss. Explain clearly in what manner provision is made to allow the pitch to be varied to a limited extent.

72. Make a rough sketch of a feathering paddle-wheel. What are the advantages of this kind of wheel as compared with the common radial wheel? How are the floats actuated?

73. How is the thrust of the screw propeller prevented from coming on the crank-shaft, and in what different ways is the thrust taken when going astern?

74. What means are provided for taking the thrust of the shaft both inboard and outboard? Sketch an ordinary thrust-block.

75. The engines of a ship develop 1,000 I.H.P. The pitch of the screw is 18 feet and revolutions per minute 100. The total area of the bearing surface of the collars is 800 square inches. Find the thrust per square inch upon them.

76. What bearings are provided for the screw shafting at its aftermost part? How are such bearings lubricated, and how is the water prevented from getting into the ship? Give a sketch.

77. Show how the bearings of a screw propeller, arranged for being hoisted up, are fitted. How is the thrust taken when going astern? What precautions should be taken to prevent undue wear in the bearings?

78. What are the advantages obtained by the employment of a feathering screw as compared with a fixed screw? In what class of ships are feathering screws fitted?

79. A ship steams at the rate of 11 knots per hour when the engines develop 1,000 I.H.P.; at what rate would she steam for 850 I.H.P.

80. A vessel is capable of steaming 2,500 knots at a mean speed of 11 knots per hour, the engines developing 3,200 I.H.P. The consumption of coal per I.H.P. per hour is 2·75 pounds. How far would she be able to steam when developing 1,900 I.H.P. on a consumption of 2·4 pounds of coal per I.H.P. per hour? The I.H.P. may be taken to vary as the cube of the speed.

81. Enumerate and describe the whole of the steam pumping power available in a first-class armour-clad ship for clearing the ship of water.

82. What information respecting the working of the slide-valve of a steam engine can be derived from examination of an ordinary indicator diagram? What is the principal object of taking indicator diagrams? Why, and to what extent, does the vacuum shown by the diagram generally differ from that shown by the vacuum gauge on the condenser?

83. Sketch and explain any arrangement with which you are acquainted for converting compound engines having two or more cylinders into simple engines. State clearly the reasons for and against the adoption of such a plan.

84. Sketch and explain the diagrams showing the action of the steam in two-cylinder compound engines with the cranks at right angles, and with a receiver between the cylinders, 1st when the cut-off is before half-stroke; 2nd when the cut-off is after half-stroke.

85. What is the effect of a fall in pressure during the exhaust from the high-pressure cylinder to the receiver? Where would you cut off in the low-pressure cylinder to reduce this fall to zero, neglecting clearance, and how would the clearance in the low-pressure cylinder affect this?

86. Explain fully how the clearance spaces in the cylinders affect the action of the steam and the efficiency of the engine, in the case of compound and simple expansion engines.

87. The high- and low-pressure cylinders of a compound engine are 50 and 100 inches diameter respectively. Stroke 8 feet and number of revolutions per minute 95. Steam pressure in the boilers 100 pounds. Vacuum 27 inches. Steam is cut off in the high-pressure cylinder at half-stroke, and in the low-pressure cylinder at $\frac{5}{8}$ stroke; draw the diagrams you would expect to obtain from each cylinder, and calculate the I.H.P.

88. Give a short description of the usual method of proceeding in raising steam, starting, and getting under way with a set of high-pressure boilers and compound engines.

89. Within ordinary limits the consumption of coal may be taken to vary directly with the I.H.P. At the maximum speeds the consumption is generally increased by the forcing of the fires, steam expended in producing the draught, &c. For practical purposes, however,

if p_m = mean effective pressure
N = number of revolutions per minute

I.H.P. may be taken to vary as N^3, or as $p_m^{\frac{3}{2}}$

From the given data, if 2,500 I.H.P. be required to drive a ship at her maximum speed of 16 knots per hour, what horse-power would be required to propel her at the rate of 8 knots? If the consumption of coal at the maximum speed be 2·3 pounds per I.H.P. per hour, and at the reduced speed, 1·9 pounds, how far would the ship be capable of steaming at full and half speed respectively, with a coal stowage of 1,000 tons? At what speed would the ship steam with half power, the slip being taken as constant, and how far would her coal carry her at this speed?

90. Suppose a ship when working at her full power of 1,200 I.H.P. to attain a speed of 12 knots with a consumption of 2·1 pounds of coal per I.H.P. per hour, and that her most economical rate of speed is 7 knots per hour. If her coal stowage is 450 tons, and the consumption per I.H.P. per hour at the most economical speed is 1·9 pounds, how far could she steam at the maximum and reduced rates of speed respectively? If the effective mean pressure calculated from the diagrams at the maximum speed be 25 pounds per square inch, what would you expect it to be at the most economical rate of speed.

91. Assuming the engines of a ship to require the same quantity of steam per I.H.P. at all speeds, and that with 3,000 I.H.P., she steams 16 knots an hour with a consumption of 54 cwts. of coal, how far will she be capable of steaming with 1,000 tons of coal at the rates of 16, 12, 10, 8, and 6 knots respectively?

92. If the coal stowage of a ship capable of steaming 16 knots per hour with 3,000 I.H.P. is sufficient to carry her a distance of 5,000 knots, how much coal would she expend in steaming the distance at 10 and 7 knots respectively? If the consumption at the maximum speed be 2·2 pounds per I.H.P. per hour, what is the quantity of coal carried? How many knots would she run per ton of coal at the several rates of speed?

93. Suppose the maximum speed of a ship to be 16 knots an hour, the engines at the time developing 4,000 I.H.P., with a consumption of 2·3 pounds per I.H.P. per hour. The coal stowage is 1,200 tons. What would be the consumption per day, and how far could the ship steam at 16, 14, 12, 10, and 8 knots respectively? What would be the consumption of coal, and how long would be required for the ship to make a passage from Portsmouth to Alexandria, a distance of 3,000 knots, at the several speeds mentioned?

94. If the travel of a slide-valve be 8¾ inches, angular advance of the eccentric 35°, outside lap 2¼ inches, and inside lap ¼ inch; find, by means of Zeuner's valve diagram, the maximum opening of the cylinder port to steam, the lead of the slide-valve, and the part of the stroke at which the steam is cut off.

95. Suppose the slide-valve to cut off the steam at $\frac{2}{3}$rds-stroke, and the maximum opening of the port to steam to be $1\frac{1}{2}$ inches, find the travel, lap, lead, &c., of the slide-valve.

96. A slide-valve that has 2 inches outside lap is set with a lead of $\frac{5}{16}$ inch, and the release at each end of the cylinder takes place at $\frac{9}{10}$ths of the stroke. What will be the difference in the amounts of inside lap at the two sides of the valve, if the length of the connecting-rod be twice that of the stroke?

97. In the case of a new design, if the average displacement and midship section coefficients (see page 416) of ships of similar form be 250 and 650 respectively, what I.H.P. would you estimate to be required to drive a ship, whose displacement is 2,500 tons and immersed midship section 480 square feet, at a speed of 15 knots per hour?

98. If the maximum diameter of the screw propeller for the ship referred to in question 97 be 16 feet and the pitch be taken 1·2 times the diameter, what number of revolutions per minute would be necessary to give the required speed, assuming a slip of 15 per cent.? If the speed of piston be 600 feet per minute, what should be the length of stroke?

99. Make a sketch, with figured dimensions, of the piston-rod for a direct-acting engine with cylinders 60 inches in diameter. Pressure of steam in the boilers 60 pounds per square inch. If the piston be fitted with two rods, of what diameter would you make them?

100. Of what dimensions would you make the connecting-rod for the engine referred to in question 99? If the crank brasses are secured by two bolts, what should be their diameter if the working strength be taken at 9,000 pounds per square inch? Make a sketch of the crank end of a rod, taking the diameter of the crank-pin to be 14 inches.

101. If this engine, when working at the full power of 2,500 horses, make 90 revolutions per minute, what should be the diameter of the propeller shafting? Sketch a flange coupling suitable for this shafting, and mark the principal dimensions on it.

102. If the distance between the centre-lines of the cylinders be 10 feet, and the length of stroke 3 feet 4 inches, make a sketch of the crank-shaft, marking on it the diameter of shaft and dimensions of the crank-arms. The shaft should be made in two symmetrical parts connected together by a flange coupling.

103. If the pressure of steam in the boilers be 60 pounds per square inch, and the total ratio of expansion be eight times, what should be the diameter of the low-pressure cylinder of a compound engine to satisfy the conditions of question 98? The back pressure

may be taken at 4 pounds per square inch. If the volume of the high-pressure cylinder be $\frac{1}{3}$rd that of the low-pressure cylinder, and the point of cut-off in the low-pressure cylinder be at $\frac{5}{8}$ths of the stroke, draw the probable indicator diagrams and calculate the power that would be developed in each cylinder.

The following approximate rules may be used in solving the foregoing questions.

If D = diameter of cylinder, in inches.
N = number of revolutions per minute.
p = initial absolute pressure of steam, in pounds per square inch.
d = diameter of piston-rod, in inches.
d^1 = diameter of screwed end of piston-rod.

Single piston-rods, $d = \frac{D}{65}\sqrt{p}$; $d^1 = \frac{D}{85}\sqrt{p}$

Double piston-rods, $d = \frac{D}{90}\sqrt{p}$; $d^1 = \frac{D}{120}\sqrt{p}$

Connecting-rod, diameter at centre $= \frac{D}{60}\sqrt{p}$

diameter at necks $= \frac{D}{65}\sqrt{p}$

diameter of bolts in flange couplings of shafts

$$= \frac{0\cdot5 \text{ to } 0\cdot55}{\sqrt{n}} \times \text{diam. of shaft}$$

where n = number of bolts in the coupling.

Crank and Propeller Shafts.

For solid shafts, diameter $= c\sqrt[3]{\frac{\text{I H P}}{\text{N}}}$

Where c = a constant.

For wrought-iron crank-shafts		$c = 4\cdot5$
„ steel	„	$c = 4\cdot2$
„ wrought-iron propeller-shafts		$c = 4\cdot0$
„ steel	„	$c = 3\cdot7$

For hollow shafts. If d_1 = external diameter,
d_2 = internal diameter,
d = diameter of the equivalent solid shaft,

then, $d^3 = \frac{d^4_1 - d^4_2}{d_1}$.

If b = width and h = depth of each crank-arm
$b\,h^2 = (\text{diameter})^3$

Mean absolute pressure.—This may be found as follows. Let A B represent the initial absolute pressure to any given scale. Produce BA to C, making AC = $\frac{1}{4}$ A B. With C as centre draw a portion of a circle cutting the horizontal line drawn through A, at the point D. Divide AD at E in such a proportion that $\frac{AD}{DE}$ will be equal to the total ratio of expansion. Then the ordinate EF drawn at E to cut the circle will represent the *mean absolute pressure*, to the same scale that AB represents the initial pressure. To find the *mean effective pressure* the back pressure must be deducted from the mean absolute pressure thus found.

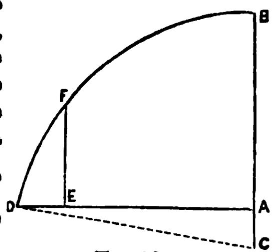

FIG. 262.

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